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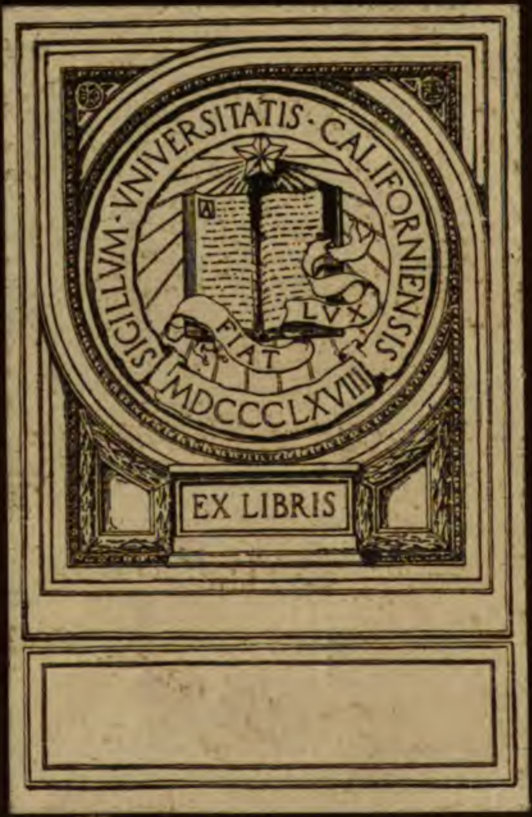

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THE GAS, PETROL, AND OIL ENGINE

THE GAS
PETROL, AND OIL ENGINE

VOL. II.

THE GAS, PETROL AND OIL ENGINE IN PRACTICE

BY

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PREFACE

THE first volume of the new edition of this work was published in 1909, so that three years have been spent in the preparation of the present volume, which deals with practical problems of design, construction and operation of Gas, Petrol, and Oil Engines. The author has been fortunate in securing the co-operation of Mr. G. A. Burls, M.Inst.C.E., in collecting information and writing the book.

Chapters I, II, V and XII are by the author; Chapters VII, VIII, IX and XI are by Mr. Burls; while Chapters III, IV, VI and X are in great part the work of Mr. Burls. The whole work, however, has been the subject of joint discussion, and the book as it now stands correctly represents the opinions of both writers.

The aim throughout all the chapters has been the comparison of early and late constructions and the discussion of various problems which have appeared in the course of the development of the various types of engine since the introduction of the 'Otto' cycle in 1876.

Chapters I and II deal broadly with the development of four and two-stroke engines, and describe various difficulties which have been encountered and partially overcome; results of operation are given and various tests are dealt with both in the works and in practical operation in factories.

Chapter III deals with igniting arrangements, and describes and discusses the advantages and disadvantages of the different methods used, tracing the development which has demonstrated the supremacy of electrical ignition in its best forms.

Chapter IV describes governors and governing methods as practised on the Continent and in England and America.

Chapter V deals with gaseous fuels, coal gas—the earliest and best of all fuels for internal combustion engines; anthracite—and coke—producer gas, pressure and suction; bituminous fuel producers, pressure and suction; coke-oven gas; and blast furnace gas.

Chapter VI discusses petroleum, petrol, and paraffin oils, and deals first with the question of motive power from oil and coal, and arrives

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at the conclusion that coal must be relied on by the world as its main source of motive power. Oil must take a secondary position. The chemical and physical properties of oil are treated from the engineer's point of view.

Chapter VII discusses and illustrates early and late petrol engines, and deals broadly with the power, weight and proportions of these engines.

Chapter VIII describes some typical recent petrol engines, and gives many particulars of the results obtained in various tests.

Chapter IX deals with the important subject of carburettors, and describes the principal types and the results obtained from them.

Chapter X describes heavy oil engines from Priestman to Diesel, and discusses the various forms of vaporiser required.

Chapter XI deals with marine oil and gas engines, including Diesel. The success of the Diesel engine in relatively large vessels has called public attention to the general problem, somewhat to the neglect of the marine gas engine.

Chapter XII deals broadly with the future of internal combustion motors and discusses some points of importance.

An Appendix deals with the question of the acceleration of the reciprocating parts.

Throughout the whole book all the practical problems within the knowledge of the writers have been discussed in a manner which, it is hoped, may prove useful to engineers engaged in the design and construction of these motors. It is hoped, too, that the book may be acceptable to students and inventors as containing numerous tables of data giving the results of many years of practice.

The writers are greatly indebted to the Institution of Civil Engineers for permission to use many blocks and to the Institution of Mechanical Engineers for permission to reproduce figs. 35–39 inclusive, and fig. 42.

They are also much indebted to both Institutions for the valuable papers published in their Proceedings, from which extracts have been made.

They tender their thanks to the many engineers and engineering firms who have kindly supplied them with information as to the construction and operation of various engines. They are particularly indebted to :—

Messrs. 'The Automotor Journal'; William Beardmore & Co.; Blackstone & Co.; Bosch; The British and Colonial Aero. Synd.; Campbell; Cockerill; Crossley Bros.; The Daimler Motor Co.; de Dion, Bouton, & Co.; The Deutz Co.; The Diesel Engine Co.;

Ehrhardt & Sehmer; Fielding & Platt; Galloways, Ltd.; The Germain Motor Co.; The Griffin Eng. Co.; R. Hornsby & Co.; Koërting; The Inst. of Automobile Engineers; The Key Engineering Co.; The Lanchester Motor Co.; Lodge Bros.; The Maudslay Motor Co.; Mather & Platt; Mirrlees, Bickerton, & Day; The National Co.; Norris & Henty; The Nuremberg Co.; Petters Ltd.; J. Pollock, Sons, & Co.; Richardson, Westgarth & Co.; Ruston, Proctor, & Co.; The Snow Steam Pump Co.; John I. Thornycroft & Co.; The Westinghouse Co.; White & Poppe; and The Wolseley Motor Car Co.

And to the following gentlemen:—Mr. P. Allen; Mr. Alan E. L. Chorlton; Mr. A. G. Ionides; Prof. Hopkinson; Mr. L. H. Pomeroy; Mr. P. A. Poppe; Mr. A. A. Remington; and Mr. J. E. Thornycroft.

They wish also to acknowledge the assistance they have derived from the work of the following authors and writers of papers on Internal Combustion Engines:—

Messrs. Percy Allen; Leonard Andrews; James Atkinson; G. H. Baillie; F. R. S. Bircham; B. Blount; R. A. Brewer; Prof. Burstall; Alan E. L. Chorlton; Cecil A. Cochrane; Herr Dubbel; Frank Foster; H. Guldner; H. Haeder; E. M. Hann; H. A. Humphrey; F. E. Junge; Ernest Koërting; R. E. Mathot; Dr. Nicolson; Sir B. Redwood; Prof. William Robinson; Messrs. Stokes and Cunningham; Prof. W. C. Unwin; John Westgarth; Prof. A. Witz; and Prof. W. Watson and colleagues.

D. C.
G. A. B.

57, 58 Lincoln's Inn Fields,
London.
November 1912.

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THE GAS, PETROL, AND OIL ENGINE IN PRACTICE

CHAPTER I

THE DEVELOPMENT OF FOUR-STROKE OR OTTO CYCLE GAS ENGINES

THE test of use and time has proved the Otto cycle to deserve a position of paramount importance; accordingly its mechanical development and present position shall be discussed first.

Four-stroke engines are now built of many types and powers, from the tiny two-inch cylinder of the motor bicycle to the huge 51-inch cylinder of the Cockerill blast furnace gas engine, and engines of all the continental modifications and dimensions are now constructed in Britain.

It is interesting and important to trace the steps of this great development, and this will best be done for England by the study of early, intermediate, and recent designs of Otto cycle engines constructed by such leading gas engine builders as the firms of Crossley Bros., Ltd., of Manchester, and the National Gas Engine Co., Ltd., of Ashton-under-Lyne, as these two firms turn out from their shops between them more than one hundred engines weekly, ranging in horse-power from one to fifteen hundred.

As Messrs. Crossleys, Ltd., are the oldest of the British firms and have constructed a very large number of gas engines, the author will first consider their engines.

CROSSLEY OTTO ENGINE OF 1880

In external appearance this engine closely resembles a modern high pressure steam engine, the working parts of which are of somewhat excessive strength; its motor and only cylinder is horizontal and open ended; in it works a long trunk piston, the front end of which serves as a guide and does not enter the cylinder proper; the connecting-rod communicates between the guide and the crankshaft;

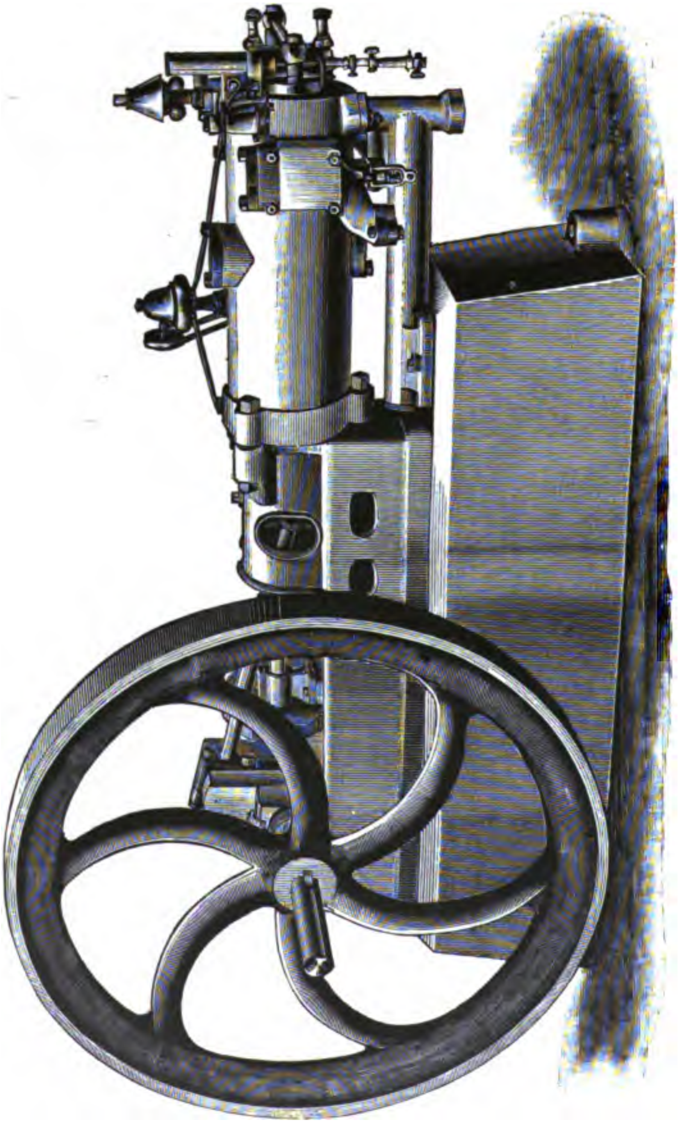

Fig. 1.—Otto Engine

the side thrust is thus kept off the piston and cylinder proper, which become hot. The crankshaft is heavy and the flywheel large, considerable energy being required to take the piston through the negative part of the cycle. The cylinder is considerably longer than

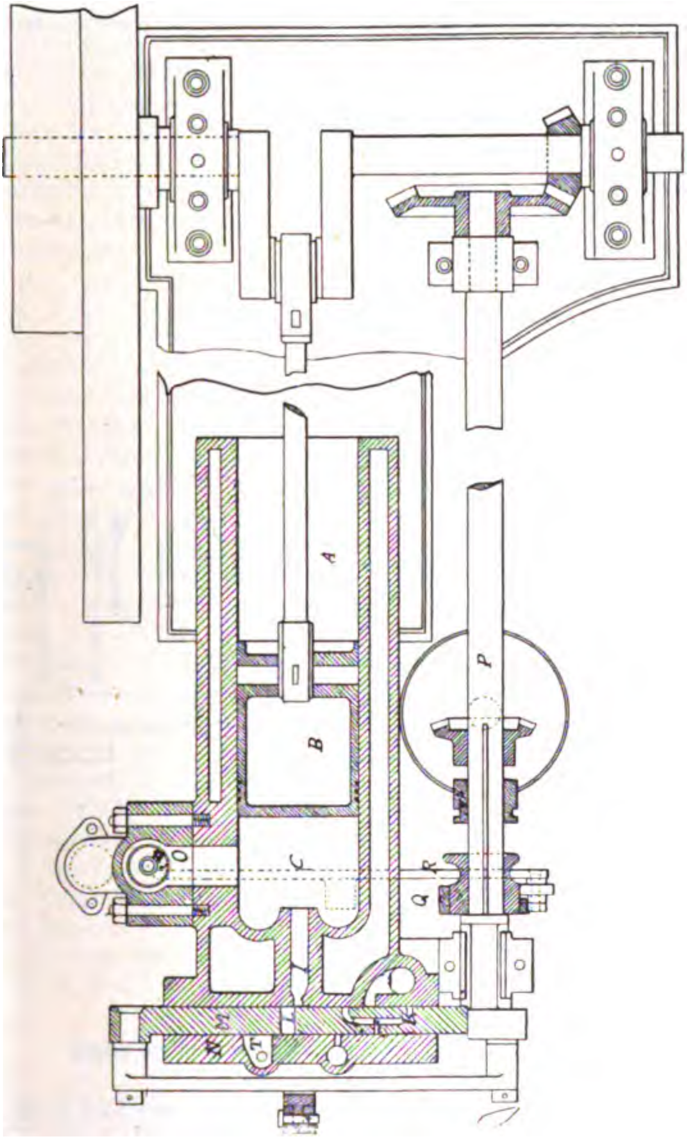

Fig. 2.—Otto Engine (sectional plan)

the piston stroke, so that the piston when full in leaves a space into which it does not enter.

Outside the cylinder, running across it at the end of the compression space, works a large slide valve ; it is held against the cylinder face by a cover plate and strong spiral springs ; it is driven to and fro by a small crank on the end of a shaft parallel to the cylinder axis,

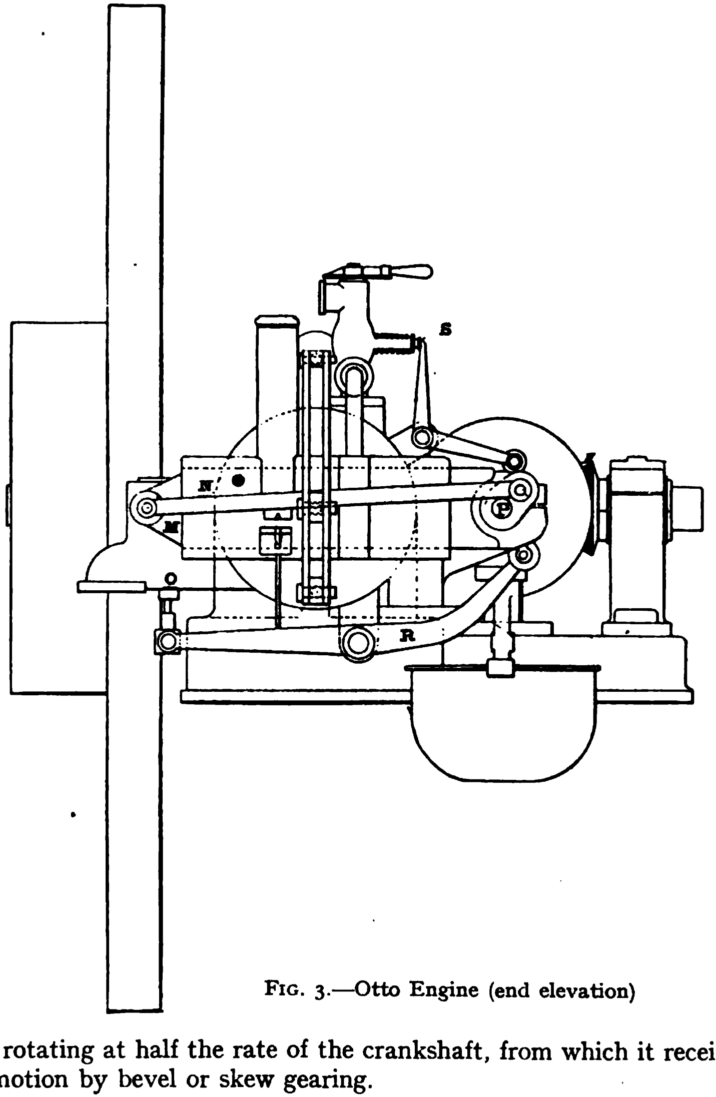

FIG. 3.—Otto Engine (end elevation)

and rotating at half the rate of the crankshaft, from which it receives its motion by bevel or skew gearing.

An exhaust valve, leading into the space by a port, is also actuated at suitable times from the secondary shaft ; the governing and oiling gear are also driven by this shaft.

The regulation of the speed of the engine is accomplished by a centrifugal governor, which is arranged to close a gas supply valve whenever the speed increases. An explosion is thereby missed, and the engine goes through its cycle as usual; but as no gas is mixed with the air, there is no explosion when the flame enters, the compressed air merely expanding, giving back to the piston the energy taken during compression.

When running without load, eight or even more revolutions may be made between the impulses; at full load two revolutions are made per impulse. Notwithstanding this irregularity, the flywheel is so large that no variation observable by the eye can be seen while watching the engine.

Fig. 1 is an external view of this Otto engine.

Fig. 2 is a sectional plan, and fig. 3 an end elevation showing exhaust valve lever.[1] A is the water-jacketed cylinder, B the piston shown full in, C is the compression space, I the admission and ignition port, communicating alternately with the gas and air admission port K and the flame port L in the slide M; N is the cover holding the slide to the cylinder face and carrying in it the external flame for lighting the movable one in flame port L. The exhaust valve is of the conical seated lift type and is seen at O; it is driven from the shaft P by the cam Q and the lever R. The other details are clearly shown upon the drawing. The ignition valve and governing arrangement will be described in a subsequent chapter; here it is sufficient to state that the governor withdraws a cam actuating the gas valve S, fig. 3, and so prevents it opening when the piston is taking in air. When open, the gas passes the valve, then through a row of holes in the valve port K, streaming into the air and mixing thoroughly with it as it enters the cylinder. To start the engine, the flame at T is lighted; the cock commanding the internal flame being properly adjusted, and the gas turned on, a couple of turns at the flywheel should cause ignition and set the engine in motion. The larger engines are provided with a second cam, which keeps the exhaust valve open during half of the compression stroke, and so diminishes the work required to turn round the engine by hand. When the engine is started the wheel upon the lever is shifted to the normal cam, and the compression then returns to its usual intensity.

Diagrams and Gas Consumption of Early Engines

The two most complete trials of Otto cycle engines at an early date are those of Dr. Slaby of Berlin and the late Professor Thurston of the Stevens Institute of Technology, Hoboken, U.S.A. Dr. Slaby's trial of a four horse-power engine was made at the Gas Motoren Fabrik Deutz in August 1881, and Professor Thurston's at Hoboken in 1884.

---

[1] Figs. 2 and 3 are sections of a later engine than Fig. 1 in which the front slide or crosshead of the piston is dispensed with.

The following are the main particulars of Dr. Slaby's tests as discussed by the author in the early editions of this book :—

The dimensions of the engine are :

Diameter of cylinder	171·9 millimetres
Stroke	340 ,,
Compression space	4770 cub. centimetres
Volume displaced by piston	7888 ,, ,,

The compression space is therefore 0·6 of the volume displaced by the piston. The results are briefly as follows :

Average revolutions during test	156·7 per min.
Power indicated in cylinder	5·04 horse
Power by dynamometer	4·4 ,,
Gas consumed in one hour	142·67 cub. ft.
Gas consumed in one hour by igniting flames	2·75 ,,
Gas consumption per IHP per hour	28·3 ,,
Gas consumption per effective HP hour	32·4 ,,

The composition of the gas used at the Gas Motoren Fabrik Deutz is given as :

Marsh gas, CH_4	34·4 volumes
Ethylene, C_2H_4	3·5 ,,
Hydrogen, H	56·9 ,,
Carbonic oxide, CO	5·2 ,,
	100·0 ,,

and 1 cubic metre of it weighs 0·404 kilogram. One pound weight of it therefore measures 39·6 cub. ft. Deducting the latent heat of steam produced, one pound weight evolves heat enough to raise 12,094 lbs. of water through one degree Centigrade. It evolves 12,094 Centigrade heat units (548 B.Th.U. per cub. ft. lower value). From this value and the experimental determination of the heat leaving the engine by way of the water jacket, Dr. Slaby calculates the disposition of 100 heat units given to the engine as follows :

	Per cent.
Work indicated in cylinder	16·0
Heat lost to cylinder walls	51·0
Heat carried away by exhaust	31·0
Heat lost from engine by conduction and radiation	2·0
	100·0

The actual indicated efficiency of the engine is therefore 16 per cent. or 0·16.

The temperature of the gases expelled during the exhaust stroke was determined by carefully protecting the exhaust pipe from loss of heat by non-conducting material, and then seeing whether zinc or antimony would melt in it. Zinc melted, but antimony did not; as the melting-point of zinc is 423° C., and the antimony melting-point is 432° C., the temperature of the exhaust gases (according to Dr. Slaby) is given with great accuracy as between these two temperatures. The average composition of the mixture is given as 1 volume of coal gas to 13·73 volumes of air and other gases. Here Dr. Slaby is plainly in error, as his own figures conclusively show. The volume of coal gas taken into the engine at each stroke as measured by the gas meter is given as 859 cubic centimetres; the total volume swept by the piston of the engine per stroke is 7888 cubic centimetres; the volume of the compression space 4770 cubic centimetres. Now, if the gas be introduced into the cylinder while it is filled completely, space included, with cold gases, at the same temperature as the gases when measured by the meter, this figure is correct enough. But the gases are not so introduced—the space is already filled with exhaust gases at a temperature of about 400° C. by Dr. Slaby's own determination; this volume must therefore be calculated to atmospheric temperature before an approach to the true ratio can be obtained. Taking atmospheric temperature at 17° C., then 4770 cubic centimetres of burned gases at 400° C. become reduced to 2055 cubic centimetres at 17° C.; that is, the total charge will consist of 859 cubic centimetres of coal gas, 7029 cubic centimetres of air, and 2055 cubic centimetres of burned gases from the previous explosion.

The ratio is:

$$\frac{\text{coal gas}}{\text{air and burned gases}} = \frac{859}{7029 + 2055} = \frac{1}{10{\cdot}5}.$$

The composition of the charge is more correctly represented as 1 volume of gas to 10·5 volumes of air and other gases. Even here, however, the dilution is overstated, as it is assumed that the piston has taken in the charge at full atmospheric temperature and pressure. But there is some throttling in passing through the admission valve and port, and also some heating of the air by striking the piston and cylinder walls. Professor Thurston, in experiments to be described later on, proves this to be the case, and shows that the charge is even stronger than has been calculated.

The temperature 400° C., it is important to note, must necessarily be lower than the temperature of the burned gases in the cylinder just previous to release, as great heat is lost in passing through the exhaust valve casing, and probably 400° C. is too low also because of the heat loss necessarily incurred in the exhaust pipe, notwithstanding its protection by non-conducting material.

It has been already proved[1] that in this type of engine, expanding after compression and explosion to the same volume as existed before compression, the theoretic efficiency is independent of the temperature of the explosion or the temperature existing before compression, and depends only upon the volume before and after complete compression. As the ratio of compression space to volume swept by the piston is 0·6 to 1, the volume before compression is 1·6, volume after compression 0·6.

The theoretic efficiency is $E = 1 - \left(\frac{v_c}{v_o}\right)^{\gamma-1}$, where v_c is the compression volume, and v_o the volume before compression; in this case $E = 1 - \left(\frac{0·6}{1·6}\right)^{0·408}$, or $1 - \left(\frac{1}{2·66}\right)^{0·408}$; hence $E = 0·33$.

That is, if all the heat were given to the engine at the moment of complete explosion at the beginning of the stroke, and no heat were lost to the cylinder during the expansion to the original volume, then 33 per cent. of that heat would be converted into indicated work.

The efficiency relatively to the air standard is therefore $\frac{16}{33} = 0·48$.

The mechanical efficiency of the engine is high, the ratio of dynamometric to indicated power being 87 to 100, and the friction of the engine only 0·64 horse-power.

Dr. Slaby's experiments are exceedingly complete, but the late Professor Thurston in America has made even more extended measurements.

Messrs. Brooks and Steward made trials under the direction of Professor Thurston, at the Stevens Institute of Technology, Hoboken. The dimensions of the engine were as follows:

Diameter of cylinder	8·5 ins.
Stroke	14·0 „

Capacity of compression space 38 per cent. of total cylinder volume.

Not only was the gas entering the engine measured, but at the same time the air required was measured through a 300-light meter. So far as the author is aware, this is the only set of early experiments in which this was done; it is by far the most accurate way of getting the true proportions of the explosive mixture

The temperature of the exhaust was measured by a pyrometer, and the power determined both by indicator and dynamometer; at the same time the heat passing into the walls of the cylinder was determined by measuring the water heated and estimating the loss by radiation and conduction.

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[1] See vol. i.

The total number of revolutions during the various tests was taken by a counter. Many trials were made under varying conditions of load and mixture used. The following is the best full power trial, giving the most economical results:

Average revolutions during test	158 per minute
Power indicated in cylinder	9·6 horse
Power by dynamometer	8·1 „
Gas consumed in one hour	235 cub. ft.
Gas consumption per IHP per hour	24·5 „
Gas consumption per effective HP per hour	29·1 „

An analysis of the gas used during the trials made by Professor Thomas B. Stillman, Ph.D., is as follows:

	Per cent.
Hydrogen, H	39·5
Marsh gas, CH_4	37·3
Nitrogen, N	8·2
Heavy hydrocarbons, C_2H_6, &c.	6·6
Carbonic oxide, CO	4·3
Oxygen, O	1·4
Water vapour and impurities (H_2O, CO_2, H_2S)	2·7
	100·0

One cubic metre of this gas weighs 0·606 kilogram. One pound weight of it, therefore, measures 26·43 cub. ft. One pound when completely burned evolves heat enough to raise 9070 lbs. of water through 1° C. (618 B.Th.U. per cub. ft. lower value).

The air necessary to supply just enough oxygen for the complete combustion of 1 volume of this gas is 5·94 volumes.

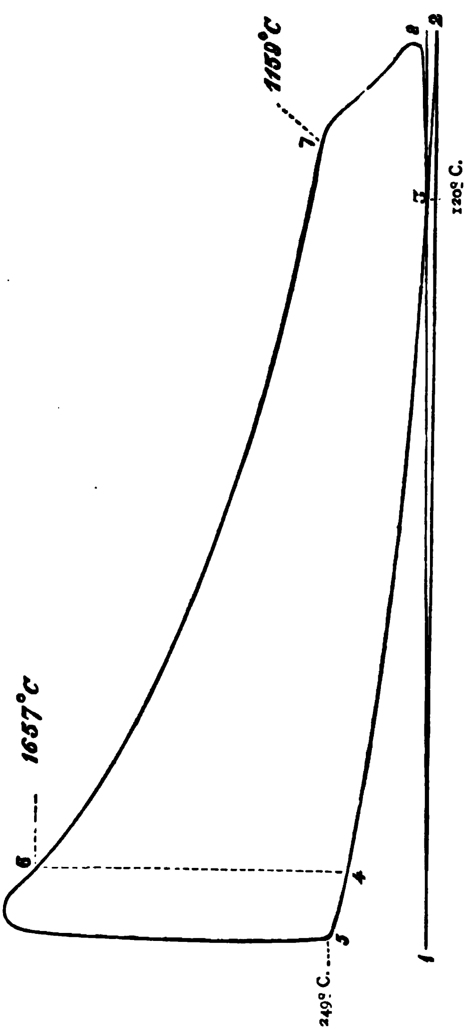
From these values and experiments upon the temperature of the exhaust gases, Professor Thurston estimates the disposition of 100 heat units by the engine as follows:

	Per cent.
Work indicated in cylinder	17·0
Heat lost to cylinder walls	52·0
Heat carried away by exhaust gases	15·5
Heat lost from engine by conduction and radiation	15·5
	100·0

The actual indicated efficiency is therefore 17 per cent.

The number showing the proportion of heat passing into the water jacket is also very nearly Slaby's, but the amount expelled with the exhaust is much understated. The amount lost by radiation is overstated.

The temperature of the exhaust gases, as determined by a pyrometer placed in the exhaust pipe, varied in the experiments at full load from 399° C. to 432° C., thus practically coinciding with Slaby. The ratio of air to gas was found, by actual measurement of both, to



8" dia. cylinder. 14" stroke. 160 revs. per min.

Fig. 4.—Normal Indicator Diagram. From 6 HP Otto Engine. (*Thurston*)

be about 7 to 1 when the engine was working most economically. Although with better gas the ratio would be slightly increased, yet it could not equal that usually given for the Otto engine, 10 to 1 or thereabouts.

The ratio is commonly obtained from a measurement of the gas consumption alone, the air being reckoned as the volume of the piston displacement, less the measured amount of gas. This is not an accurate method, for the reason already stated.

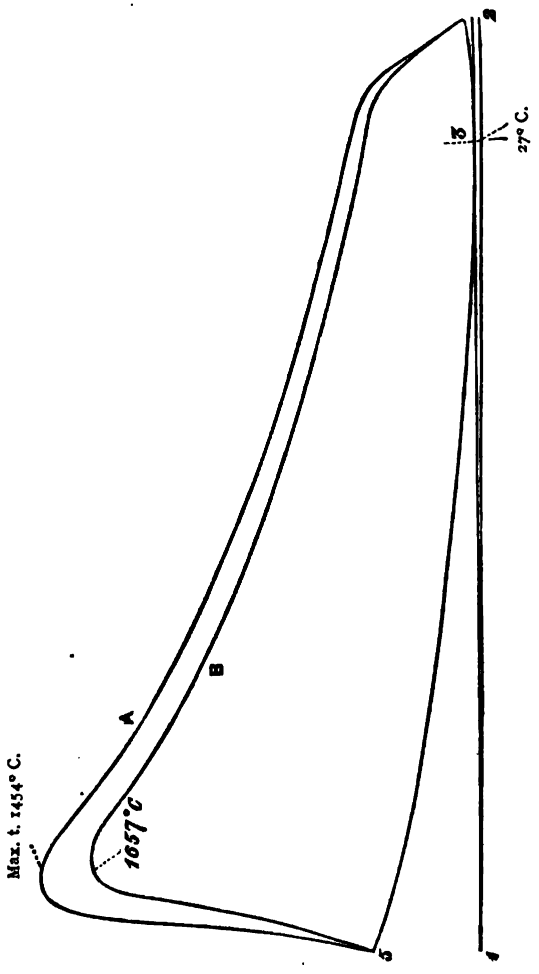


FIG. 5.—Diagram with regular and with intermitted action. 6 HP Otto Engine. (*Thurston*)

If the mixture filling the cylinder mingles with the burned gases filling the compression space, then the average composition of the charge is 1 volume of coal gas to 9·1 volumes of other gases.

Fig. 4 is a fair sample of the diagrams obtained during Professor Thurston's tests while the engine was giving full power. The piston while moving from the point 1 to the point 2 takes in the charge; the pressure in the cylinder falls below atmosphere as the piston approaches the end of its stroke. This is due to the resistance of the valve port to the entering air and gas. The piston returns from 2 to 5 (first in-stroke), compressing the charge, the pressure increasing to atmosphere at the point 3, compression being complete at the point 5; ignition then occurs, and the pressure and temperature rapidly rise as the explosion progresses; the temperature does not attain its maximum till the piston has moved forward a little and has reached the point 6. From that point to 7, when the exhaust valve opens, the expanding line is as nearly as possible apparently adiabatic. The temperatures are marked at each point of the diagram. The return stroke from 8 to 1 discharges the products of combustion. This is the second in-stroke, completing the cycle and leaving the engine in position to again take in a fresh charge.

The diagram shows the whole changes occurring during two complete revolutions of the machine while fully loaded. Fig. 5 shows what occurs when the governor acts, when the engine is at less than full load. The smaller diagram, B, is the normal one, and the larger, A, the intermittent one; the gas has been completely cut off for several strokes, and so the hot burned gases in the compression space have been completely discharged and replaced by pure air at a temperature not far removed from atmospheric; the explosion then causes a higher pressure by nearly half an atmosphere, although the maximum temperature is less than in the usual case.

The temperature of the charge before explosion being less, a smaller increase is required to produce a given increase of pressure. Professor Thurston calculated that the heat accounted for by the diagram was 60 per cent. of the total heat supplied to the engine; the deficiency he attributed to the phenomena of dissociation, which prevented the complete evolution of the heat at the highest temperature, but permitted further combustion when the temperature fell. The amount of gas required to run at full speed, 166 revolutions per minute without any load, was found to be from 50 to 70 cub. ft. per hour.

Other things being equal, better results are obtained with large engines. The theoretic efficiency is constant for both large and small engines where the same compression is in use, but the loss of heat from the explosion to the sides of the cylinder is less in large engines, due to the diminished surface exposed in proportion to the volume used. The effect is to increase the efficiency of the gas in the mixture used, a smaller quantity being necessary to make up for the loss of heat.

The indicator diagrams prove the very efficient nature of the Otto cycle. The great simplicity attained by the alternate use of the cylinder as pump and motor diminishes the number of valves necessary, and secures the minimum resistance to the entering gases, while entirely preventing any loss due to ports in transferring the gases from one cylinder to another. The carrying out of the cycle is mechanically almost perfect, little work being spent which is not included in the theory. Again, the piston is full in at the moment of ignition and is almost at rest; the heat, producing maximum temperture, is therefore added at nearly constant volume. The highest pressure which the gas present is capable of producing is therefore attained at the beginning of the stroke simultaneously with the highest temperature; the succeeding expansion is then very rapid, and so no unnecessary waste of heat occurs, the temperature being rapidly depressed by work being done. The united effect of all the arrangements is seen in a diagram which is very perfect.

The losses incurred in the operation of the cycle have already been fully discussed in the first volume.

CROSSLEY OTTO ENGINE BUILT IN 1892

Careful drawings were made by the author in 1895 of a Crossley Otto engine of 9 HP (nominal) built in 1892, and now at work at the Clifton Rocks Railway, Bristol. The engine is numbered 19772. It has been thought best to select an actual engine as an example, in order to clearly appreciate the points of difference from the earlier engines. The particular engine selected was tested by the author for power and gas consumption. The engine shows many points of advance over the early engines, but, curiously enough, although it possesses all the necessary valve arrangements to enable high compression pressures to be utilised, yet defects in the proportion of the compression space and piston prevented the use of high compression, and the engine did not give the best economy possible for the particular type. Accordingly the gas consumed per brake HP hour was 25·9 cub. ft. This is a much better result than would have been obtained from a slide valve Otto engine such as illustrated at figs. 1, 2, and 3 of this volume, but it is not nearly so good as the type allows.

Fig. 6 is a side elevation of the engine; fig. 7 is a plan part in section; fig. 8 an end elevation; figs. 9 to 13 inclusive are drawn to a larger scale; fig. 9 is a side elevation of the back end of the cylinder looking on the cam shaft; fig. 10 is a corresponding plan; fig. 11 an end elevation; fig. 12 a vertical longitudinal section through the cylinder; and fig. 13 is a separate section on a still larger scale of the igniter tube and funnel.

A comparison of the illustrations with those of the earlier slide valve engine at once shows great mechanical development and points of constructive difference. Thus in the early engine the crosshead guide and the engine cylinder were two distinct parts, requiring to be

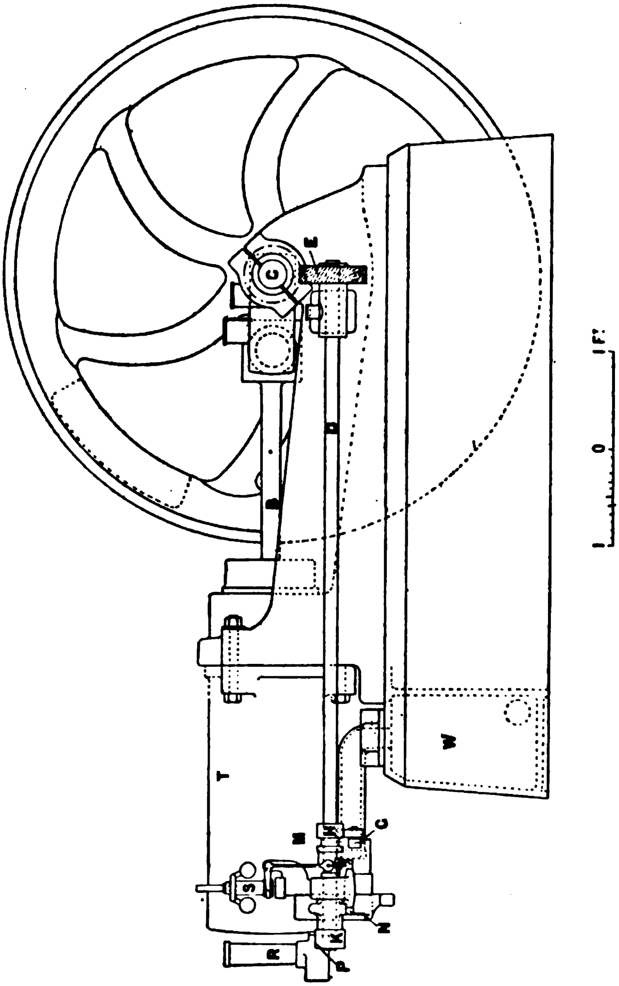


Fig. 6.—Crossley Otto Engine, 9 HP Nominal (elevation)

bolted together in accurate alignment in order to allow the piston with its crosshead slide to work freely without jamming ; in the later engine a long trunk piston is used which serves the double purpose of piston and crosshead guide ; the separate crosshead slide is, in fact, dispensed with, and consequently the cylinder serves as its own slide guide, requiring no adjustment of separate parts. The cylinder, that is, serves both as cylinder and slide guide, and the whole cylinder is bolted to the bed against a powerful faced flange.

The bevel wheels of the early design are also dispensed with, and replaced by skew or worm wheels, which besides taking up much less space provide a much quieter drive for the two to one shaft. The unsightly distortion of the bed shown in fig. 2 necessary to admit the bevel wheels is quite avoided, as is clearly seen at fig. 7. There are many smaller points of constructive difference which the experience of years has shown to be desirable, but the great points of departure are to be found in the suppression of the flame slide valve method of ignition, and the introduction of the incandescent tube igniter; the diminution of the relative volume of the compression space, which is not carried out to its proper extent in this individual engine; and the improved proportioning of the valves and ports in order to minimise throttling of the charge during the inlet period, and back pressure of the exhaust gases during discharge.

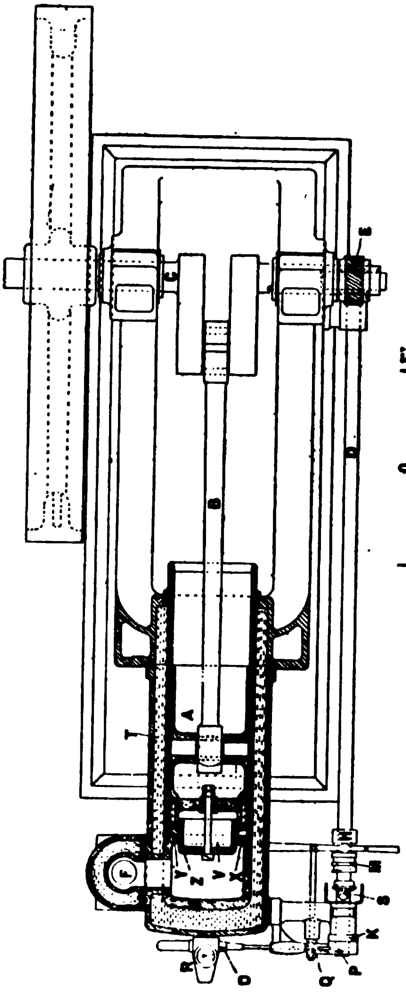


FIG. 7.—Crossley Otto Engine, 9 HP Nominal (plan part in section)

The engine follows the same cycle of operations as the old engine; that is, by one movement of the piston it takes into the cylinder a charge of gas and air, which is compressed on the return stroke into a space at the end of the cylinder, there to be ignited in order to give the explosion and produce the power stroke; the power stroke is then followed by the exhausting stroke, and the engine is ready to go through the same operations to prepare for another power stroke. In this engine the charge of gas and air is admitted by the inlet valve I, which is of the conical seated lift type; the valve is operated by the lever J from a cam, K, on the valve shaft D. This valve shaft is rotated at half the speed of the crankshaft by means of worm wheels or skew gear, E. The gas supply is admitted to the inlet valve I by the lift valve L, which valve is also operated by the lever and link N and cam M, controlled, however, by the centrifugal governor S. The governor operates either to admit gas wholly or cut it off completely, so that the variation in power is obtained by varying the number of the explosions. The exhaust valve F is also a conical seated lift valve, and it is actuated by the lever C and cam H. The ignition is produced by admitting a portion of the compressed inflammable charge from the compression space to the tube R, rendered incandescent by the Bunsen flame. The passage to the igniter tube

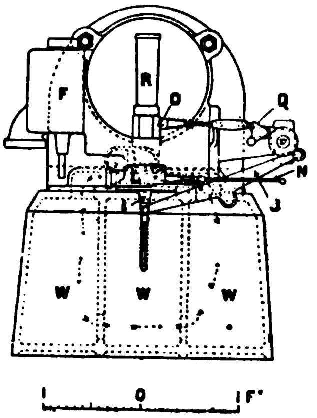


FIG. 8.—Crossley Otto Engine, 9 HP Nominal (end elevation)

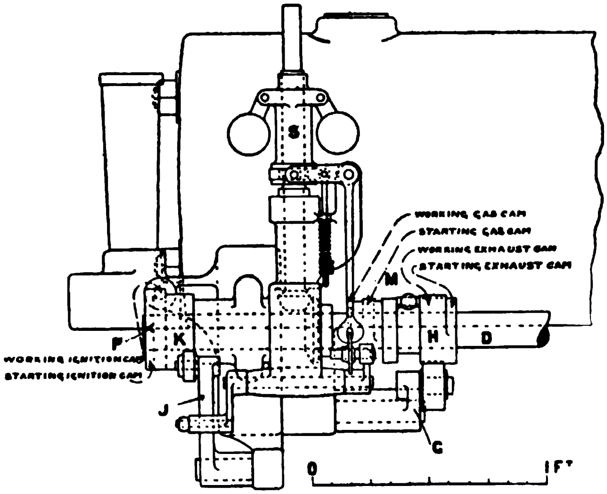


FIG. 9.—Crossley Otto Engine, 9 HP Nominal (side elevation, back end)

is controlled by the valve O, which valve is operated by the lever Q and cam P. The valve O is double seated, and during the compression period of the engine the face nearest the compression space is kept up against the seat by a powerful spring ; the incandescent tube is thus kept open to the atmosphere, and notwithstanding any leak which may occur from the cylinder the tube remains empty until the moment when it is required for ignition. When the valve is lifted from one seat a small portion of the compressed mixture is discharged through a small port to the air, and this clears out the burned gases which would otherwise render ignition irregular, and permits pure combustible mixture to reach the incandescent internal surface of the tube when the outer valve face closes on its seat. This device causes the ignition of the explosive mixture at the proper time.

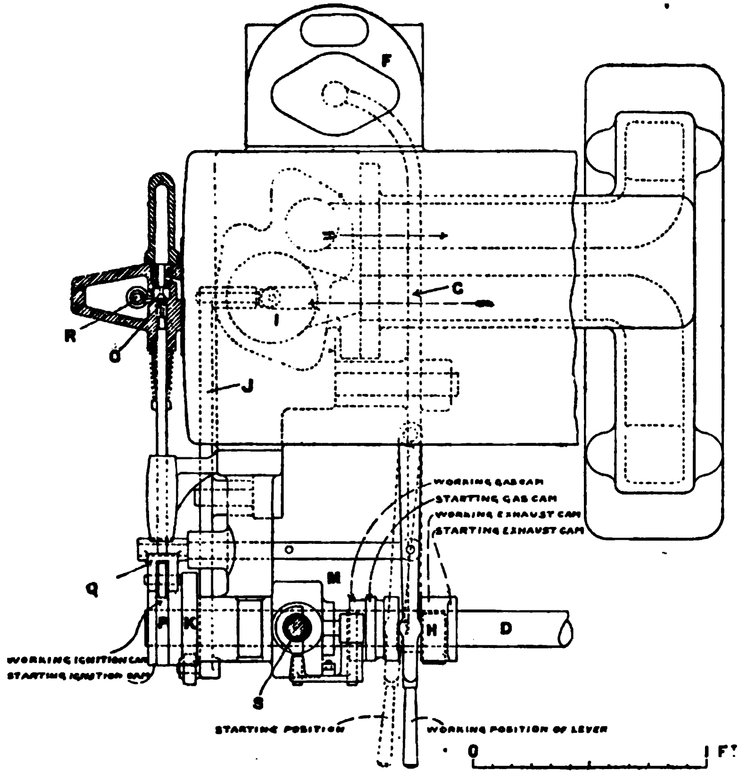


FIG. 10.—Crossley Otto Engine, 9 HP Nominal (plan, back end)

The adoption of lift valves for the admission and discharge of gases to and from the engine cylinder simplifies the practical problem of admitting and discharging with the least possible throttling or wire drawing. So long as slide valves were used to admit the charge to the cylinder, it was difficult to provide a sufficiently large inlet area, as the area allowed in a port bearing against a slide surface determined the pressure necessary to hold the slide against the valve face to prevent the escape of flame when the compressed mixture was exploded. In a six horse-power engine of the early type, for example, the inlet port in the back cover was 2⅜ ins. long by ⅝ in. wide, equal to 1·5 sq. ins. Assume the maximum pressure of the explosion to be 150 lbs. per sq. in., then the slide valve must be pressed to its working face with a pressure not less than 225 lbs.; as a matter of fact the slide was pressed up to its work with a pressure of about 600 lbs. When it is considered that the flame temperature during the explosion is about 1600° C., it is easy to comprehend the difficulty of keeping the slide cool enough to maintain a good working surface even at comparatively low pressures. Designers of slide engines for this reason were forced to content themselves both with the minimum of port area and with low compressions. Small port area produced, naturally, considerable

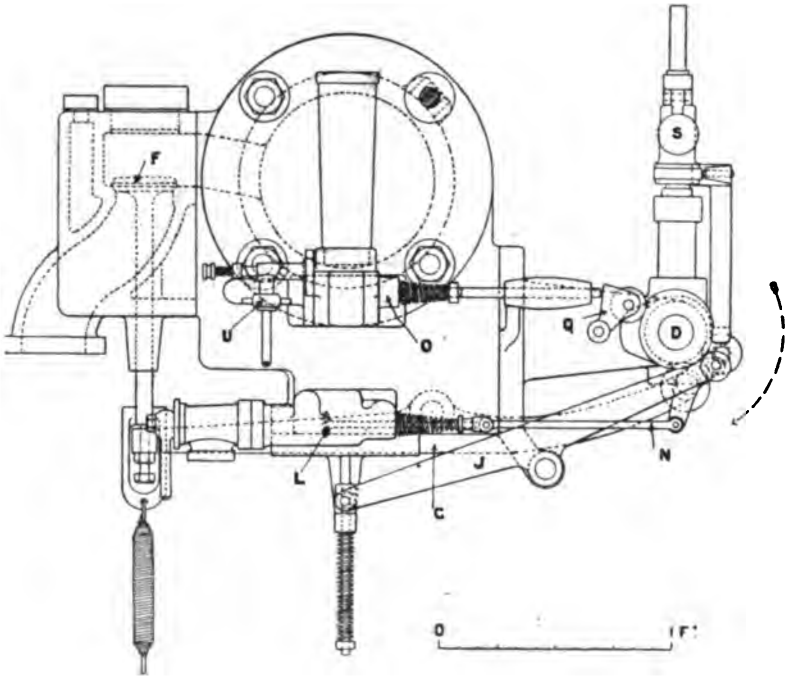


FIG. 11.—Crossley Otto Engine, 9 HP Nominal (end elevation)

resistance to the inflowing charge, and low compressions prevented the attainment of any great economy of gas consumption. In the old engines the velocity of flow of the air and gases entering the cylinder often exceeded 244 ft. per second, so that when the piston reached the out end of its stroke the cylinder was not filled up to atmospheric pressure. The evil of throttling in this way was not confined to the positive loss of power due to the resistance to the charging stroke of the piston; the greatest loss was caused by the considerable reduction in the weight of the charge drawn in, and the consequent increase in the proportion of the exhaust gas present. In many cases it was found that the contents of the cylinder were at a pressure of 1½ lbs. per sq. in. below atmosphere when the engine terminated its charging stroke, and this meant that the total volume of charge admitted was reduced by 20 per cent. as compared with the charge which would have entered had the admission area been sufficient to allow the cylinder to fill up to atmospheric pressure. The proportion is greater because of the large volume of the compression space which must be allowed for in calculating the loss due to defect of pressure. The slide valve was undoubtedly a formidable difficulty in these engines, now happily

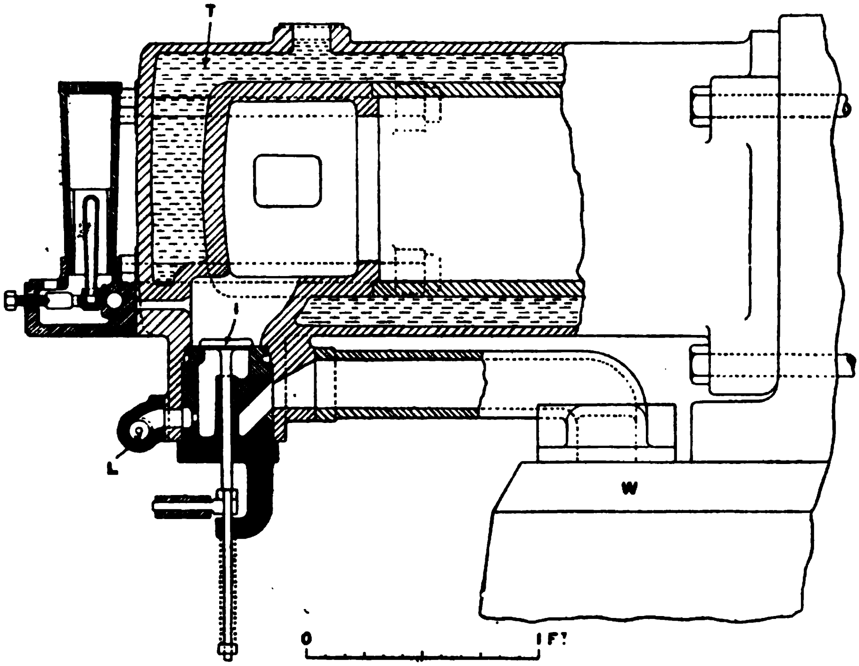


FIG. 12.—Crossley Otto Engine, 9 HP Nominal (vertical section, end)

overcome by the substitution of lift valves. With lift valves it is easy to provide any desired admission port area, as the pressure of the explosion holds the valve to its seat, and large valves may be used just as readily as small ones. In the engine illustrated in figs. 6 to 13 the admission area is 4·3 sq. ins. with the valve full open, and, assuming maximum opening to remain during the whole charging stroke, the velocity of the entering charge is only 130 ft. per second. This engine is therefore better supplied with combustible mixture than the old slide engine.

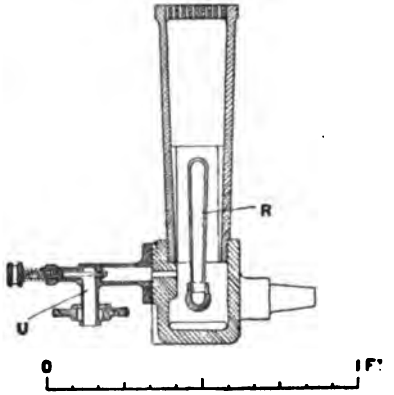


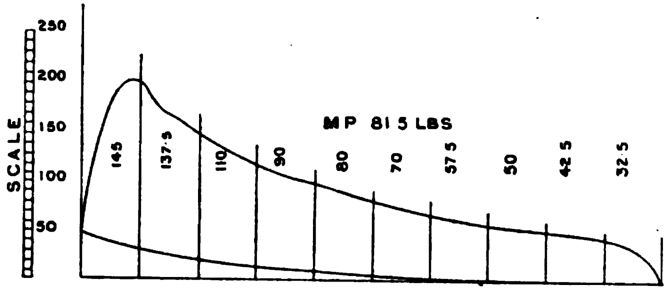
Fig. 13.—Crossley Otto Engine, 9 HP Nominal (section of tube igniter)

The compression pressure in a slide valve engine is limited by the difficulty of preventing a slide from cutting on its face at high compression and explosion pressures, and this difficulty is also overcome by the use of lift valves when combined with an incandescent tube igniter.

In the older engines the importance of a free exhaust exit was not fully recognised, and although the exhaust valves were lift valves, the discharge area provided was insufficient. Thus in the six-horse slide valve engine referred to, the average velocity of the exhaust gases past the exhaust valves was 137 ft. per second; in this engine it is only 80 ft. per second. The exhaust gases are thus better discharged in the later engine. Any increase in the volume of the exhaust products causes loss of economy in a gas engine; a small proportion does little harm, but a large volume of exhaust heats the entering charge and so raises the temperature of compression. Premature ignitions are also caused by the compression of a charge mixed with hot exhaust. Designers now endeavour to expel exhaust products as completely as possible.

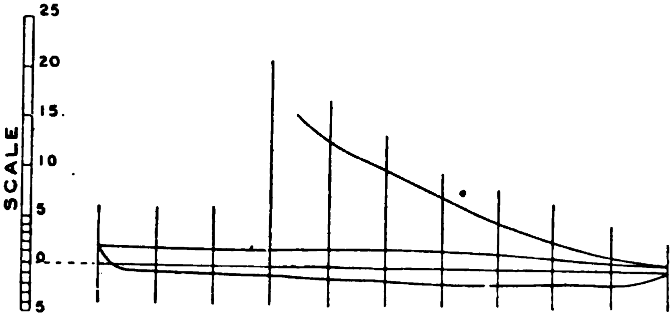
The engine illustrated has several bad points, and it appears to the author to be one issued by the makers while they were in a transition stage, probably engaged in increasing their compression pressures. To get the best possible results from a given volume of explosive mixture, it should be compressed into a combustion space, having the minimum of port capacity communicating with the admission and exhaust valves. In the older engines this point was not appreciated, and the port capacity was always excessive. In this engine the port capacity back to the exhaust and inlet valves is undoubtedly too great. Ports act as condensers for the flame of the explosion, and rapidly cool the ignited charge at a time when it least bears cooling.

Any narrow spaces should also be avoided, and this engine presents an example of attempting to increase compression, by means of the block V attached to the piston, which should be carefully avoided. It will be noticed that the block V, fig. 7, projects into the combustion space through the reduced diameter part X, and so forms the annular space Y between the piston proper and the reduced casing. This annulus has a cooling effect on the flame under the explosion pressure while the piston A is practically stationary, but it has a much more serious cooling effect whenever the piston begins to move out. The flame gases then pass through the space between the piston block V and the ring X into the annulus Y, and so the flame is dragged through a cooling or condensing surface, and considerable loss is thus caused. Indeed it may be at once stated that, to gain the greatest advantage from high compressions, the whole of the compressed explosive mixture



Nominal HP, 9; diam. of cylinder, 9½″; length of stroke, 18″; revs. per min., 160; indicated HP, 19·25; consumpt. per IHP per hour, 21·2 cub. ft.; consumpt. light, 70 cub. ft. per hour; brake HP, 15·75; consumpt. per BHP per hour, 25·9 cub. ft.; mean pressure, 31·5 lbs.; max. pressure, 200 lbs.; pressure before ignition, 46 lbs.; scale of spring, $\frac{1}{120}$″ per lb.

FIG 14.—Crossley Otto Engine, 9 HP Nominal (full load diagram)



Scale of spring, $\frac{1}{16}$″ per lb.; mean pressure, 2·5 lbs.; charging resistance, 0·7 IHP; total resistance running light, 3·3 IHP.

FIG. 15.—Crossley Otto Engine, 9 HP Nominal (light spring diagram)

should be contained in *one* space, that is, a space which is not divided into smaller separate spaces. Ports should be avoided if possible, and the flame should never be caused to flow through a narrow space into a wider one, as is done in this engine. The compression space should in fact be as nearly cubical or spherical as possible. Notwithstanding these defects, the engine shown in the illustration gives much better results than the old slide valve engines. For the purpose of comparison the author made practically simultaneous tests on the engine illustrated and on an old slide valve engine of six horse-power (nominal). The results obtained are given in the following table, and all the important valve settings and numbers are also given.

PRINCIPAL PARTICULARS OF A 6 NHP CROSSLEY OTTO GAS ENGINE, BUILT ABOUT 1881, AND A 9 NHP CROSSLEY OTTO GAS ENGINE, No. 19772, BUILT IN 1892.

—	6 NHP Engine No. 4683. 6" diam. cylinder × 18" stroke	9 NHP Engine No. 19772. 9¼" diam. cylinder × 18" stroke
Volume swept by piston	804 cub. ins.	1275·8 cub. ins.
Volume of compression space	516 cub. ins.	510 cub. ins.
$\frac{\text{Vol. swept by piston}}{\text{Vol. of comp. space}}$	$\frac{804}{516} = \frac{1}{0{\cdot}64}$	$\frac{1275{\cdot}8}{510} = \frac{1}{0{\cdot}4}$
Compression pressure	31 lbs. per sq. in. above atmosphere.	48 lbs. per sq. in. above atmosphere.
Explosion pressure	125 lbs. per sq. in. above atmosphere.	200 lbs. per sq. in. above atmosphere.
Mean available pressure	57 lbs. per sq. in.	81·5 lbs. per sq. in.
Revolutions per min.	164	160
Indicated horse-power	9·0	19·25
Brake horse-power	6·75	15·75
Gas consumption per hour (including ignition)	236 cub. ft.	408 cub. ft.
Gas per IHP per hour	25·5 cub. ft.	21·2 cub. ft.
Gas per BHP per hour	34 cub. ft.	25·9 cub. ft.
Mechanical efficiency	75 per cent.	81 per cent.
Area of charge inlet port	(Slide valve) 1·5 sq. in.	Inlet valve 2⅜" diam. × ⅞" lift.
Inlet port setting	Is ¼" open when piston is on in centre, and ⅛" open when piston is on out centre.	Opens dead on in centre, is held open on out centre, and closes when the piston returns 1½" in. At 1" in movement of piston the valve is $\frac{5}{16}$" open.
Exhaust valve	(2¼" diam. × ⅜" lift) 2·65 sq. in. area.	3" diam. × 1¼" lift
Exhaust valve setting	Opens while piston is 1" in from out end of stroke. Closes when piston has crossed in centre and moved out ¼".	Opens while piston is 2¼" from out end of stroke. Closes exactly on in centre.

Principal Particulars of a 6 NHP Crossley Otto Gas Engine, &c.—*Cont.*

—	6 NHP Engine No. 4683. 6" diam. cylinder × 18" stroke	9 NHP Engine No. 19772. 9½" diam. cylinder × 18" stroke
Ignition lead . . .	Ignition port in slide is ⅛" open when crank is on in centre.	(Lift valve tube igniter) valve $\frac{5}{16}$" diam. × $\frac{5}{16}$" lift, opens 1½" before compression is complete, but only full open ½" before compression is complete.
Charge velocity . .	244 ft. per sec.	130 ft. per sec.
Exhaust velocity . .	137 ft. per sec.	80 ft. per sec.
Piston speed . . .	437 ft. per min.	480 ft. per min.
Power absorbed charging and exhausting	0·7 IHP	0·7 IHP
Gas inlet valve . . .	⅜" diam. × ⅜" lift.	1" diam. × ⅜" lift.
Gas inlet valve setting .	When piston has made 1¼" forward stroke valve opens.	When piston has gone 2¼" forward stroke valve opens, and does not close till out centre has been crossed and piston returns 1¼". Valve is $\frac{3}{16}$" open when piston is full out.

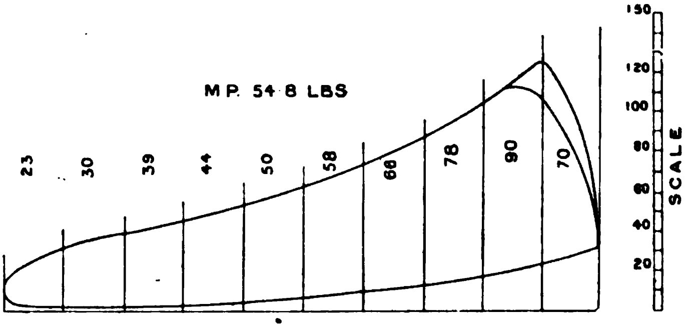
Fig. 14 is a diagram from the engine illustrated. It is a fair example of those taken during the test.

Fig. 15 is the corresponding light spring diagram.

Fig. 16 is a diagram from the slide valve engine which has been referred to; and Fig. 17 is a light spring diagram also from the slide valve engine.

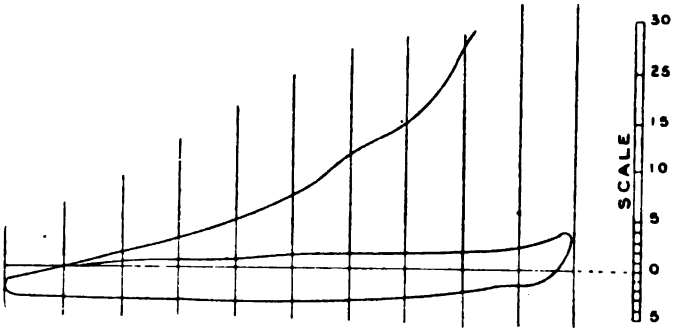
The scales of the diagram figs. 14 and 16 are different, as one required a much stronger indicator spring. It will be observed that the slide valve engine only gives an available working pressure of 54·8 lbs. per sq. in., while the lift valve engine gives 81·5 lbs.; and on comparing the light spring diagrams it will be seen that with the slide valve engine the pressure falls considerably below atmosphere at the end of the charging stroke, while with the other engine the pressure rises nearly to atmosphere before the stroke terminates.

The Crossley Otto engines built in 1880 differ to a considerable extent from the engine No. 19772 which has been here discussed. Figs. 18 and 19 show the external appearance of these engines. Fig. 18 shows the 30 HP nominal engine of 17 ins. cylinder and 24 ins. stroke, intended for ordinary driving and running at 160 revolutions per minute. Fig. 19 is the 30 HP nominal electric lighting engine of 17 ins. diameter cylinder and 21 ins. stroke, which runs at 230 revolutions per minute, and with coal gas will indicate a maximum power of 117 horse. The engines then supplied were of the 'scavenging' type. The general external appearance is similar to that illustrated, but an important modification is made in the operations performed by the engine. In addition to the cycle of operations described, the engine is so arranged that the exhaust gases formerly remaining in the combustion space are swept out and the combustion space filled with air. The combustible charge in this engine is therefore a pure mixture of gas and air without any exhaust gases. To accomplish this clearing out of the burned gases and their replacement by air, advantage is taken of the oscillations or waves of pressure set up in the exhaust pipe by



Nominal HP, 6; diam. of cylinder, 8″; length of stroke, 16″; revs. per min., 164; indicated HP, 9; consumpt. per IHP per hour, 25·5 cub. ft.; brake HP, 6·75; consumpt. per BHP per hour, 34 cub. ft.; mean pressure, 54·8 lbs.; max. pressure, 125 lbs.; pressure before ignition, 32 lbs.; scale of spring, $\frac{1}{80}$″ per lb.

FIG. 16.—Crossley Otto Engine, 6 HP slide valve (full load diagram)



Scale of spring, $\frac{1}{10}$″ per lb.; mean pressure, 3·85; charging resistance, 0·7 IHP; total resistance running light, 2·25 IHP.

FIG. 17.—Crossley Otto Engine, 6 HP slide valve (light spring diagram)

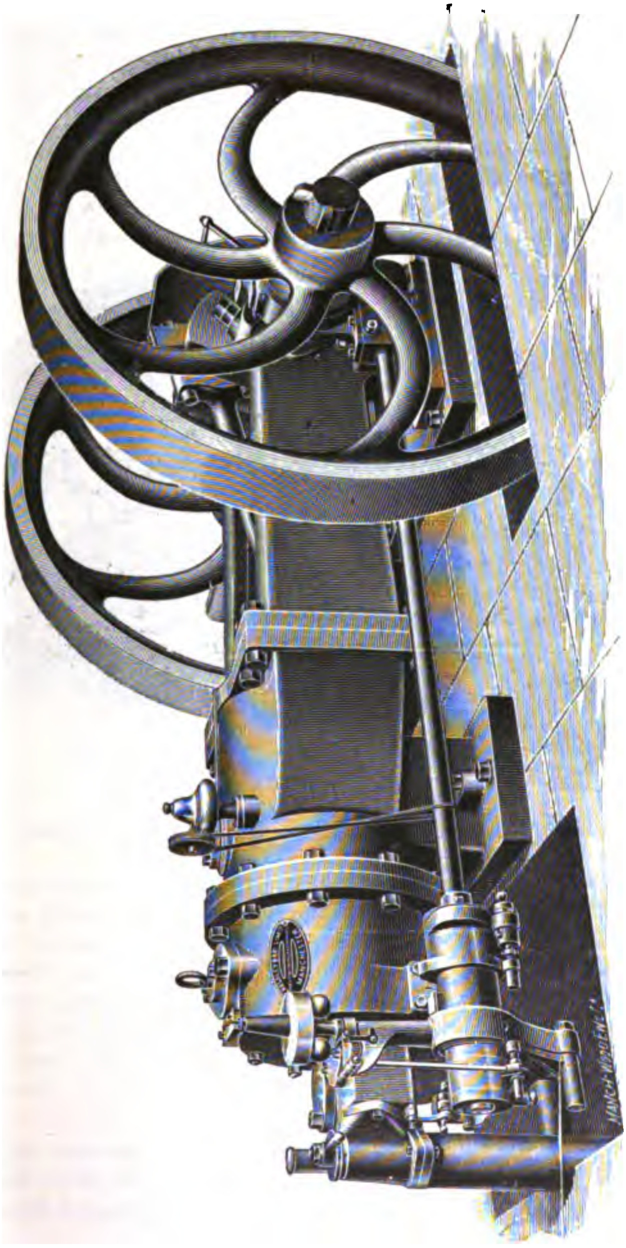


FIG. 18.—Crossley Otto Scavenging Engine, 30 HP Nominal

the discharge of the exhaust gases. It has long been known that in a gas engine exhaust pipe the pressure of discharge is succeeded by a partial vacuum, and this vacuum again succeeded by pressure, in fact that under certain circumstances an oscillation of pressure is set up in the exhaust pipe, giving a fall of pressure at certain periods after the exhaust valve is opened. Messrs. Crossley & Atkinson took advantage of this fact, and so controlled the pressure wave and the following vacuum that after the exhaust gases have been liberated from the cylinder of the engine the high pressure discharge is succeeded by a vacuum, the period of the vacuum coinciding with the approach of the piston to the end of its exhaust stroke. By then keeping open the exhaust valve and opening the charge or inlet valve while the exhaust valve is open, a charge of pure air is drawn through the combustion space to sweep out the burned gases from the compression space. When the charging stroke is complete the whole cylinder is thus filled with a pure mixture of gas and air without the deleterious burned gases. To accomplish this sweeping out in a satisfactory manner it is necessary to shape the cylinder so as to favour free flow of the entering air.

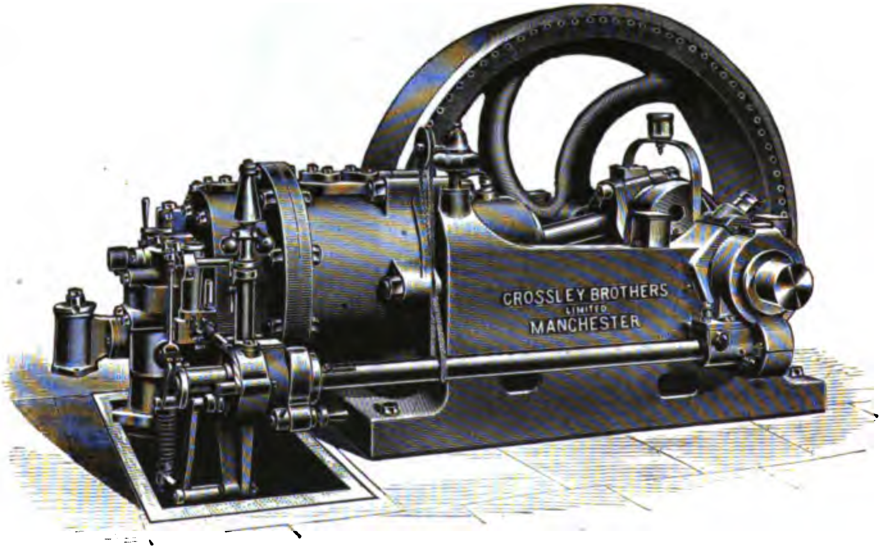


FIG. 19.—Crossley Otto Scavenging Engine, Electric Lighting, 30 HP Nominal

Figs. 20 and 21 show the arrangement of the cylinder and valves in vertical and horizontal section. The arrows in fig. 21 illustrate the flow of the scavenging air. Fig. 22 illustrates in a diagrammatic way the settings of the valves in that engine.

The desired delay in the production of the vacuum is brought about by attaching an exhaust pipe about 65 ft. long. Quieting chambers may be placed at the end of that length of pipe without affecting the result, but no large expansion chamber should be put nearer to the engine cylinder. The energy of discharge of the exhaust sets the long column of gases filling the pipe in oscillating motion, and enables a considerable reduction of pressure to be produced just as the piston is completing its exhausting stroke. Fig. 23 is a light spring diagram taken from the engine during the author's test, and it plainly

FIG. 20

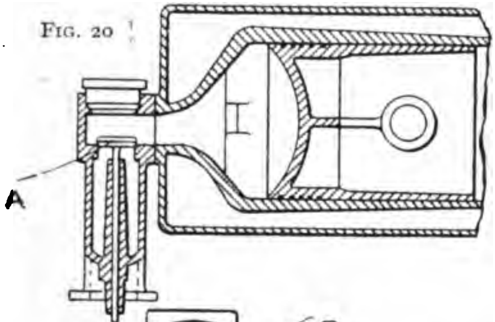
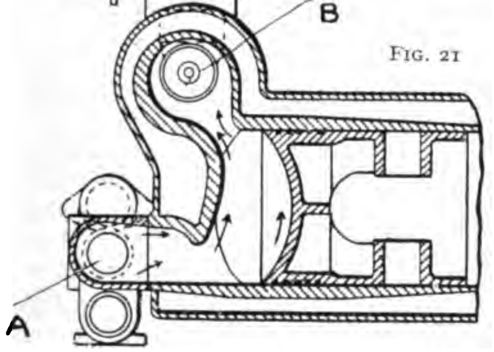


FIG. 21



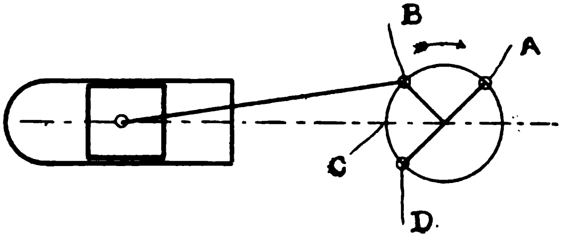
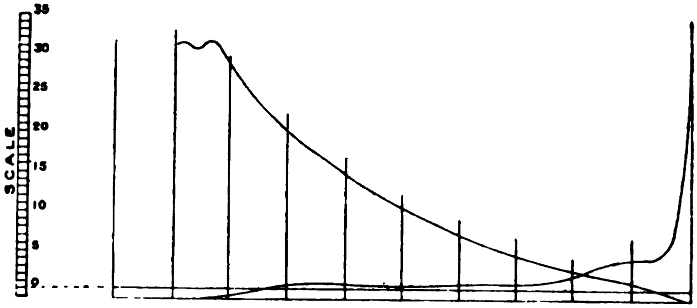


FIG. 22.—Crossley Otto Scavenging Engine (valve settings)

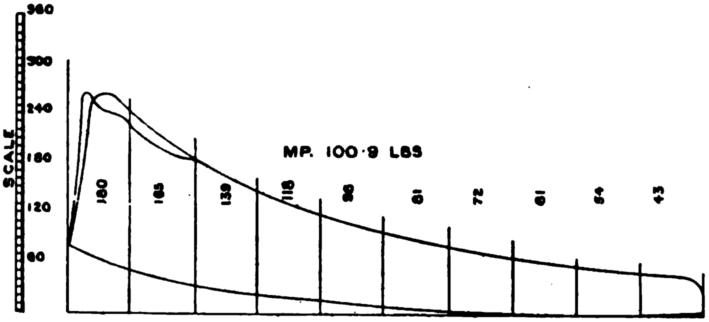
shows the effect of the vacuum so produced in the exhaust pipe. It will be noted that at the termination of the exhausting stroke the pressure in the cylinder has fallen to 2 lbs. per sq. in. below atmosphere, a reduction of pressure amply sufficient to cause a flow of air from the atmosphere to sweep through the cylinder.



Scale of spring, $\frac{1}{18}$″ per lb.; charging and scavenging diagram; charging diagram of 4 NHP Crossley Otto Engine.

FIG. 23.—Crossley Otto Scavenging Engine (light spring diagram)

In fig. 22 the air inlet valve is opened while the crank is in the position D, and the exhaust valve is held open till the crank reaches the position B. The exhaust valve opens again at A after explosion, and it is held open to B position instead of as usual to C position. The inlet valve is thus held open during the existence of a partial vacuum in the exhaust pipe, and so a 'scavenging' charge of air is drawn through the combustion space and the products replaced by pure air.



Nominal HP, 4; diam. of cylinder, 7″; length of stroke, 15″; revs. per min., 200; indicated HP, 14; consumpt. per IHP per hour, 14·5 cub. ft.; brake HP, 11·97; consump. per BHP per hour, 17·0 cub. ft.; mean pressure, 100·9 lbs.; max. pressure, 274 lbs.; pressure before ignition, 87 lbs.; spring, $\frac{1}{180}$″.

FIG. 24.—Crossley Otto Scavenging Engine (power diagram)

Fig. 24 is a diagram taken by the author during a test by him of a 4 HP scavenging engine at Messrs. Crossley's works, Openshaw. The leading particulars are marked upon the diagram, from which it will be observed that the engine gave results which were most remarkable for the time both from the points of power and economy. The engine, although only 7 ins. diameter cylinder and 15 ins. stroke, gave practically 12 brake horse-power on a gas consumption of 17 cub. ft. per brake horse-power hour, a surprisingly good result for so small an engine. Openshaw gas is 20 candle-power, and has a heat value of 530,000 ft. lbs. per cub. ft. (686 B.Th.U. per cub. ft.).

This small scavenging engine thus shows a brake thermal efficiency of 21·9 per cent., and an indicated thermal efficiency of 24·8 per cent., with a mechanical efficiency of 85 per cent.

Diagrams and Gas Consumption.—The diagrams given at figs. 14, 15, 16, 17, 23, and 24 illustrate very fairly the progress made in the Crossley Otto engine from the old slide valve engine to the lift valve scavenging engine, and it is interesting to compare the consumption of these three engines. They are as follows:

—	Gas per IHP hour	Gas per BHP hour	Compression pressure per square inch above atmosphere
Slide valve engine . .	25·5 cub. ft.	34 cub. ft.	30 lbs.
Lift valve engine No. 19772	21·2 „	25·9 „	46 „
Lift valve scavenging engine	14·5 „	17 „	87·5 „

The advance made by Messrs. Crossley is quite unmistakable; the brake consumption was just about half of the consumption in a Crossley Otto engine built in 1881. No doubt many of their slide valve engines were more economical than the one tested by the author, and the gas consumption of engine No. 19772 does not represent the most favourable result attained by Messrs. Crossley before the advent of the scavenging engine. Thus the Crossley engine tested at the Society of Arts trials at the end of 1888 had a cylinder of 9·5 ins. diameter and a stroke of 18 ins. The gas consumed per indicated horse-power per hour was 20·55 cub. ft., and per brake horse-power 23·87 cub. ft. The compression pressure was 61·6 lbs. per sq. in. above atmosphere. The indicated power was 17·12 horse, brake power 14·74 horse, and the speed of the engine 160 revolutions per minute. The mean effective pressure was 67·9 lbs. per sq. in., and the initial pressure of the explosion 197 lbs. per sq. in. above atmosphere.

The author's test of the 4 HP Crossley Otto scavenging engine was made in August 1894, so that, taking the Society of Arts Crossley engine as the most economical up to that date, from 1888 to 1894 Messrs. Crossley succeeded in reducing the gas consumption per brake horse-power from 24 to 17 cub. ft.

It is to be remembered that this figure of 17 cub. ft. per brake horse-power was obtained with a small engine. Mr. Atkinson, of Messrs. Crossley, has given the author results of a test made with an engine of 11¼ ins. diam. cylinder and 21 ins. stroke, also at Manchester. The power indicated was 46·8 horse, and the gas consumption was only 13·55 cub. ft. per IHP hour. The consumption of 17 cub. ft. per brake HP per hour is the lowest of which the author had experience, at the date 1894, with an engine so small. It will be observed that increasing economy in the Crossley Otto engine has always been accompanied by an increase of compression; thus a compression of 30 lbs. in the slide valve engine of 1881 had been replaced in 1894 by a compression of 87·5 lbs.

Compression has evidently some part in securing the advantages of the later engine. Mr. Atkinson, in a paper read before the Manchester Association of Engineers, attributed the whole of the economy of this engine to the discharge of the burned gases and their replacement by pure air. In this the author does not agree with him.

Crossley Otto Engines to 1910

The advance in thermal efficiency made by Messrs. Crossley from 1894 will be best shown by considering the results of tests made by Mr. Humphrey, Prof. Burstall, Mr. Atkinson, Prof. Hopkinson, Dr. Nicolson, and the author. The particulars are as follows:

Indicated and Brake Thermal Efficiency of Crossley Otto Engines from 1894 to 1910

Names of experimenters	Year	Dimensions of engines	Indicated thermal efficiency	Brake thermal efficiency	Mechanical efficiency	Compression lbs. per sq. in. above atmosphere
		Dia. stroke	Per cent.	Per cent.	Per cent.	
Clerk .	1894	7″ × 15″	24·8	21·9	85	87
Humphrey	1900	26″ × 36″	31·0	25·7	83	80
Burstall .	1904	14″ × 21″	37·4	30·8	82·2	over 200
Atkinson .	1905	11·5″ × 21″	35·3	30·6	86·6	150
Hopkinson	1908	11·5″ × 21″	36·8	32·0	87·5	160
Nicolson .	1908	32″ × 26″	35·6*	31·3	88*	160

* Mechanical efficiency estimated by Mr. Atkinson as 88 per cent., and indicated efficiency calculated by him from brake as 35·6 per cent.

These figures show a considerable advance in thermal efficiency for both small and large engines. Comparing the two smaller engines as tested by Clerk in 1894 and Hopkinson in 1908, the brake thermal efficiencies are respectively 21·9 per cent. and 32 per cent., an increase of about 46 per cent. The large engines tested by Humphrey in 1900 and Nicolson in 1908 show brake thermal efficiencies of 25·7 per cent. and 31·3 per cent. respectively, or an increase of a little more than 21 per cent. The compression pressures used have been increased from 87 to 160 lbs. per sq. in. above atmosphere.

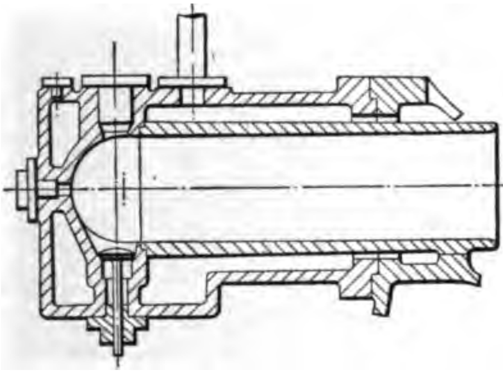


FIG. 25

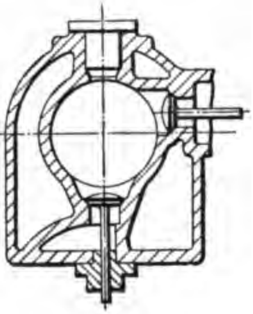


FIG. 27

To enable the changes to which the improvement is due to be followed, Messrs. Crossley have kindly supplied the author with

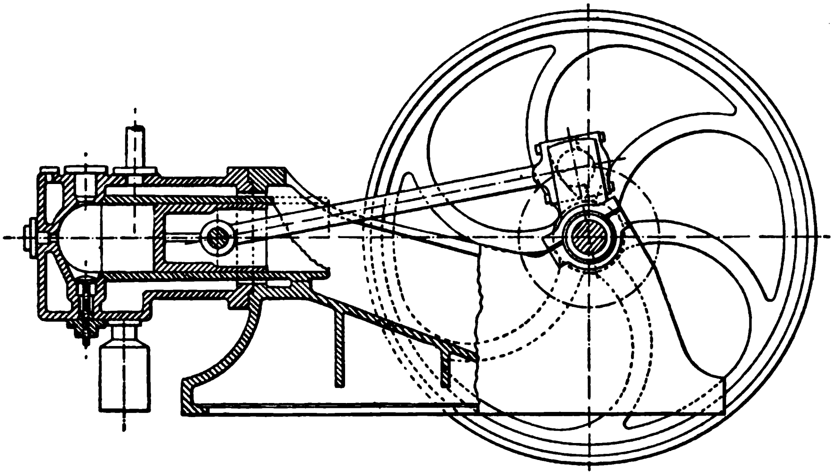


FIG. 26.

drawings of the two engines tested by Prof. Hopkinson and Dr. Nicolson.

The 40 HP Crossley engine used by Hopkinson is shown at figs. 25, 26, 27, and 28.

Fig. 25 is a longitudinal vertical section through the cylinder.

Fig. 26 is a general sectional view of the engine.

Fig. 27 is a transverse section through the combustion chamber; and

Fig. 28 is a photograph showing the general appearance of the engine.

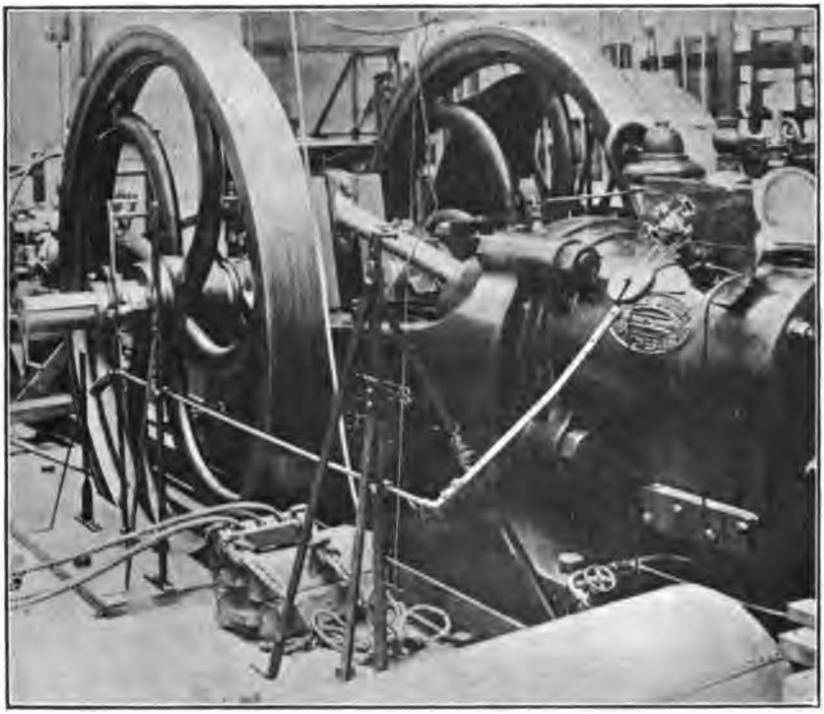


FIG. 28

The dimensions and other particulars of the engine are as follows:

Cylinder	11½ ins. diam. × 21 ins. stroke
Speed	180 revs. per minute
Compression space	407 cub. ins.
Compression ratio	6·37, that is $\frac{1}{r} = \frac{1}{6·37}$
Compression pressure	175 lbs. per sq. in. abs.
Air standard efficiency	52·2 per cent.

Ignition by magneto.

The best results obtained by Hopkinson are as follows:

Indicated horse-power = 39·3
Brake horse-power = 34·4
Mechanical losses, HP = 4·9

Mechanical losses, including charging losses and friction, as follows:

	HP
Suction (pumping loss)	1·4
Piston friction	2·5
Other frictions, valve lifting, &c.	1·1
	5·0

Mechanical efficiency	87·5 per cent.
Indicated thermal efficiency	36·8 "
$\frac{\text{Indicated efficiency}}{\text{Air standard efficiency}} = \frac{36·8}{52·2}$	70·5 "
Brake thermal efficiency	32·2 "
Mean pressure	88·5 lbs. per sq. in.
Maximum pressure of explosion	365 lbs. per sq. in. above atmosphere

$$\frac{\text{Explosions}}{\text{Cycles}} = 0·896$$

Gas used 0·1006 cub. ft. per suction at 11° C. (52° F.) and barometer 30·46 ins.

$$\frac{\text{Gas}}{\text{Total charge}} = 0·0865$$

Calorific value of gas, lower value, 570 B.Th.U. per standard cub. ft.
Water jacket temperature 88° C. (190° F.)

Hopkinson's tests were made by rope dynamometer and optical indicator. Fig 29 shows the average diagram for the above test, and figs. 30 and 31 are reproductions of optical diagrams giving respectively charging losses and release pressures.

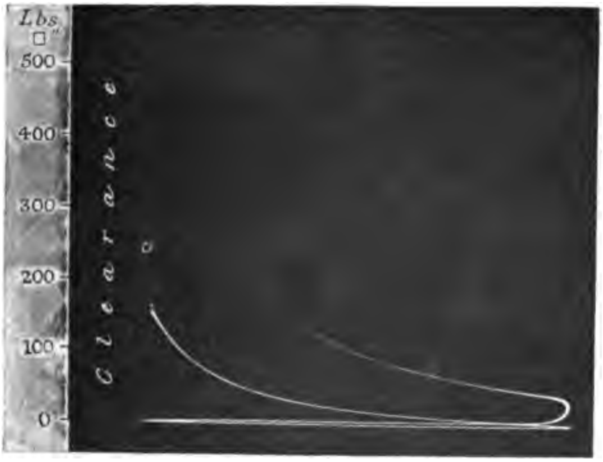


FIG. 29

In the above test the mixture used was the weakest which would fire regularly, producing a normal diagram with the explosion line

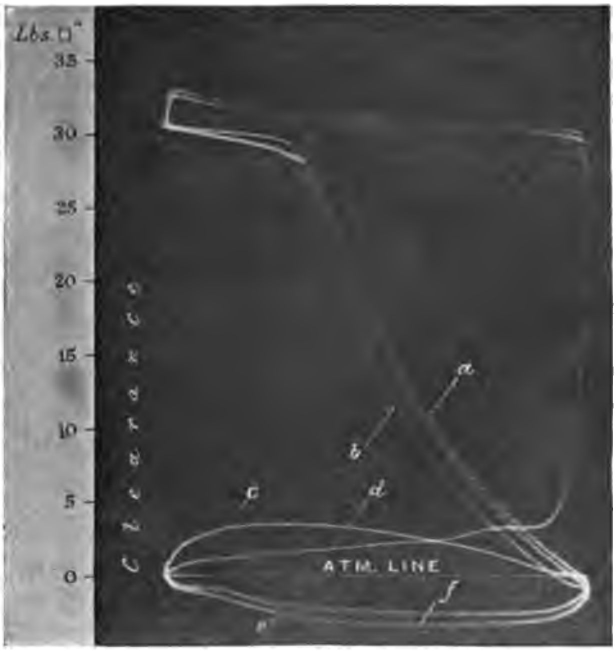

Lbs. □"
35
30
25
20
15
10
5
0
Clearance
a
b
c
d
ATM. LINE
f
e

Fig. 30

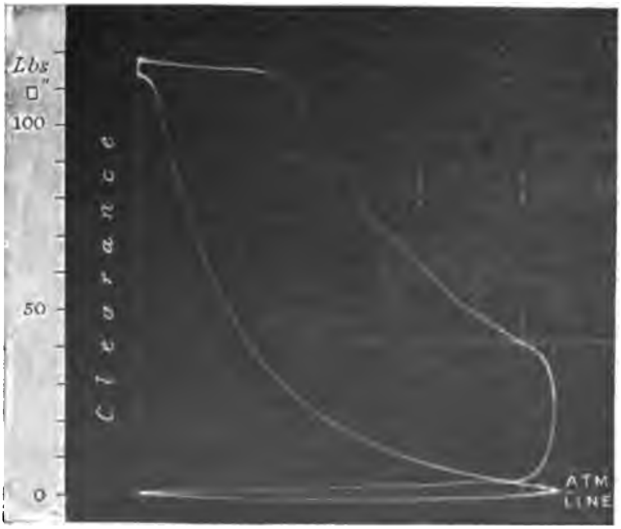

Lbs
□"
100
50
0
Clearance
ATM
LINE

Fig. 31

nearly vertical; the gas taken in per explosion was 0·1006 cub. ft. measured at the standard gas holder, equal to 0·095 cub. ft. at standard temperature and pressure; the percentage of gas in the whole cylinder contents was then found to be 8·5, or about 1 volume of coal gas to 10·8 volumes of air and exhaust gases. Calculating from the indicated thermal efficiency above 36·8 per cent., and determining the heat in the exhaust from the pressure of release as shown at fig. 31, using Holborn and Austin's values for volumetric heat, Hopkinson gives the following approximate balance sheet:

	Per cent.
Indicated work	37
Heat in exhaust (from release pressure)	42
Heat loss during expansion (by difference)	21
	100

With stronger mixtures the indicated thermal efficiency fell regularly. The strongest mixture used in his experiments contained 0·1294 cub. ft. of coal gas measured at the gas holder, equal to 0·122 standard cub. ft., or 11 per cent. by volume of the total cylinder contents, or 1 volume of coal gas to 8·1 volumes of air and other gases. This strong mixture gave the balance sheet:

	Per cent.
Indicated work	33
Heat in exhaust (from release pressure)	39
Heat loss in expansion (by difference)	28
	100

The change of strength of mixture from 8·5 per cent. to 11 per cent. of coal gas thus caused the indicated efficiency to fall from 37 per cent. to 33 per cent.

The mean pressure increases, however, from 88·5 lbs. per sq. in. with the weak mixture to 102 lbs. with the strong.

Cambridge coal gas was used throughout the tests of the following average composition:

ANALYSIS OF CAMBRIDGE COAL GAS. (*Hopkinson*)

	Percentage by volume	O required for combustion	Steam produced	CO_2 produced
H	47·2	23·6	47·2	—
CH_4	35·2	70·4	70·4	35·2
Heavy hydrocarbons .	4·8	22·6	16·0	14·4
CO	7·15	3·6	—	7·15
N	5·4	—	—	—
Other gases . .	0·25	—	—	—
	100·00	120·2	133·6	56·75

The composition of the gas varied somewhat from day to day; the higher calorific value varied from 350 Centigrade heat units (630 B.Th.U.) to 378 Centigrade heat units (680 B.Th.U.), and the lower calorific value from 317 Centigrade heat units (570 B.Th.U.) to 345 Centigrade heat units (620 B.Th.U.). The lower calorific value may thus be taken approximately as 90 per cent. of the higher.

The balance sheets given above depend entirely upon the optical indicator and on an accurate knowledge of the volumetric heats of the gases used in the engine—nitrogen, oxygen, steam, and carbonic acid mainly—at both low and high temperatures. As the volumetric heats and even the gas temperatures are not yet very accurately known, Hopkinson made twenty-five tests for heat balance with his exhaust calorimeter, using the indicator and sometimes the brake also. As there is great difficulty in determining the heat loss of an engine by radiation, he followed the Institution of Civil Engineers Committee in using the brake power instead of the indicated as an item in the balance sheet.

Three of these tests used practically the same gas as in the weak mixture balance sheet on p. 33; they are Tests 1, 7, and 33 of Hopkinson's Table of balance sheets, p. 452 of Hopkinson's Paper.[1] The gas charges were 0·093, 0·094, and 0·093 standard cub. ft. per explosion respectively, so do not differ materially from the figure of the other test.

Hopkinson gives these balance sheets as follows:

Test Number	1	7	33	Mean	—
	Per cent.	Per cent.	Per cent.	Per cent.	
Brake horse-power .	29	29	29	29·0	
Exhaust calorimeter .	40	34	35	36·3	45·6% exhaust heat
Exhaust gases . .	7	10	11	9·3	
Water jackets . .	25	28	25	26·0	
	101	101	100	100·6	

In this table the brake horse-power and the other values are calculated as percentages of the higher calorific value of the coal gas used. This is done because the exhaust gas calorimeter, in cooling down the hot gases discharged from the engine cylinder at the end of the expansion stroke, condenses the steam formed by the explosion, so that the latent heat of the steam appears as well as the heat of cooling. This, of course, reduces the apparent brake efficiency as compared with the value calculated from the lower calorific value of the gas. The water jacket heat is determined in the usual way by measuring the flow of water through the jacket and its rise of temperature between inlet and outlet. The item 'exhaust gases' is determined by means of the anemometer, giving total volume discharged from the exhaust calorimeter. The temperature of the exhaust gases leaving the calorimeter varied between 34·1° C. and 56·2° C. With regard to this item Prof. Hopkinson states, 'The quantity of gas discharged is known from anemometer measurement, probably within 3 or 4 per cent. This is saturated with water vapour when it leaves the calorimeter, and, its temperature and pressure being known, its internal energy per cubic foot can readily be calculated from steam tables. The internal energy at 15° C. can be calculated in the same way. The difference of their energies, plus the work done by the atmospheric pressure in the contraction of the gas as it cools, is equal to the heat evolved per standard cubic foot.'

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[1] Inst. M.E., April 1908.

Hopkinson points out that his balance sheets do not include radiation loss from the engine, and he describes a test made to estimate its probable value as follows:

'No satisfactory method has yet been suggested of separately determining the radiation, but some notion of its magnitude may be obtained by comparing the jacket loss at the same gas-charge with a hot and cold jacket. A number of tests agreed in showing that when the temperature of the water jacket at exit is 70° C. the heat taken away by the water is less than at 40° C. by between 100 and 150 thermal units per minute, the gas-charge and all other circumstances being the same. When the engine is fully loaded this is equivalent to between 2 and 3 per cent. of the whole supply. As the indicator diagram is not affected to any perceptible degree by the jacket temperature, the heat actually received by the engine must be nearly the same in the two cases, and the difference must be mainly due to the higher radiation at the high temperature. Since there is still some radiation of heat at 40° C. it seems probable that the total radiation at 70° C. is at least 3 per cent. The average balance sheet radiation shown in the tests at this temperature is only 1 per cent., so that there must be systematic errors in one or more of the items going to form the balance sheet. On the other hand, it is hardly possible that the aggregate of these errors can amount to so much as 3 per cent., or that the radiation at 70° C. can be more than 4 per cent.'

Hopkinson considers that he has possibly undervalued the higher calorific value of the gas used in the tests by from 1 per cent. to 2 per cent. owing to a change in the correction of the wet meter used for the calorimeter tests during the time elapsing between the first and the last.

Testing with the strong mixture charge of about 0·122 cub. ft. of standard coal gas per explosion, he found the following results, using the higher calorific value as before :

Test number	10	11	12	Mean	—
	Per cent.	Per cent.	Per cent.	Per cent.	
Brake horse-power	26	27	26	26·3	
Exhaust calorimeter	30	31	32	31·0	43% exhaust heat.
Exhaust gases	13	12	11	12·0	
Water jackets	31	29	32	30·6	
	100	99	101	99·9	

The difference between rich and weak mixtures is well shown by comparing the mean values, as below :

—	Rich mixture	Weak mixture
	Per cent.	Per cent.
Brake horse-power	26·3	29·0
Exhaust calorimeter and exhaust gases	43·0	45·6
Water jackets	30·6	26·0
	99·9	100·6

The mixture containing only 8·5 per cent. of coal gas when compared with that containing 11 per cent. shows 29 per cent. of the higher calorific value as against 26·3 per cent., but the heat passing away with the exhaust gases is greater with weak mixture than with strong, 45·8 per cent. against 42 per cent., and the reverse is the case with the water jacket heat, as with the weak mixture the water jacket carries away only 26 per cent. against 30·6 per cent. with the stronger mixture.

From figs. 25, 26, and 27 it will be seen that the combustion space is arranged so that cooling surface is kept at a minimum ; there are no ports or passages leading either to the exhaust or inlet valves. The exhaust valve opens directly into the bottom of the combustion chamber, and the charge inlet valve opens equally directly into the side. The piston has no projections and forms no annular cooling spaces such as have been referred to in the earlier Crossley engines. The gas valve, like the charging valve, is arranged horizontally, and it is controlled by the governor on the ‘ hit or miss ’ principle, so that governing is accomplished by regulating the number of impulses applied to the piston. No attempt is here made to graduate the impulses. This size of engine is ignited either by tube or low tension magneto. As shown in the photograph, fig. 28, it is fitted with hot tube ignition.

As the compression is somewhat high for use with coal gas—about 160 lbs. per sq. in. above atmosphere—the engine is fitted with a device for admitting a water spray charge during the suction stroke, and this arrangement prevents pre-ignition. Prof. Hopkinson's tests were conducted without the water spray, as he had no trouble from pre-ignition during the short full load experiments which were necessary for his purpose. In practice Messrs. Crossley use water injection when long runs are made under heavy loads.

Professor Burstall made an interesting test of a somewhat larger Crossley engine at their Manchester works on November 25, 1904, in which water injection was used for the purpose of avoiding pre-ignition. On this system Messrs. Crossley have made the following published statement:

'Our system was simply to employ high compression in conjunction with the injection of a small water spray in a particular manner which we have patented. We claim no special economy for the use of water. We find that as we use it, it prevents pre-ignition and any undue rise of temperature in the cylinder, making it possible to use gas with a high percentage of hydrogen at a very high compression. It also improves general conditions by reducing temperature through the cycle of operations.'

Professor Burstall's report is as follows:

'It has long been known that one method of improving the efficiency of gas engines is to increase the compression or, what is the same thing, to increase the ratio of expansion.

'Up to the present, when using coal or producer gas, the compression has been limited to about 110 lbs. per square inch; compressions higher than this are apt to heat up the charge to such an extent as to cause premature ignition, that is, the charge ignites before the end of the compression stroke.

'Messrs. Crossley have recently brought out a new type of engine in which the compression is carried to over 200 lbs. per square inch, and in which premature ignition is prevented by drawing in a small quantity of water during the suction stroke.

'The engine tested was of the electric lighting type with an extra heavy flywheel and an outer bearing, which to some extent accounts for the mechanical efficiency being lower than in their ordinary type of engine. The engine cylinder was 14 ins. diameter and 21 ins. stroke, the duration of the test was 6 hrs. and 45 mins. During this time the engine ran perfectly smoothly without either premature ignitions or back explosions.

'The power was taken up on an all round rope brake, water cooled; all weights, diameter of brake wheel, diameter and length of stroke of cylinder, were measured up by me at the end of the test.

'Indicator diagrams were taken every quarter of an hour, on a special Crosby gas engine indicator.

'The gas was measured on a standard meter which was certified by the Meter Testing Department of the Manchester Corporation on November 21, 1904.

'The calorific value of the gas was determined at intervals in a Junker's calorimeter, the average value being 578 B.Th.U. per cubic foot at a temperature of 60° F. and under a barometer of 30 ins.

'The clearance volume was 0·243 cub. ft., the cylinder volume being 1·872 cub. ft., and the ratio of expansion 8·7.

'The maximum pressure in the cylinder was 528 lbs. above atmospheric pressure, the volume at maximum temperature being 0·247 cub. ft.

'The average pressure during suction was 1·9 lbs. per square inch below atmosphere, that is, 12·8 lbs. per square inch absolute.

'The compression curve is given by the equation, $PV^{1\cdot268}$ equals 6,508 foot lbs., and the expansion curve by $PV^{1\cdot183}$ equals 17,610 foot lbs.

'The ratio of air to gas as found from the exhaust gases is 10·2.

'During the suction stroke the injected water appears to be turned into steam, which becomes superheated during compression; heat passes into the charge from the walls until a pressure of about 160 lbs. per square inch is reached, when the charge appears to be as hot as the walls; the heat additions prove that the whole of the available heat, that is, the total heat of the charge less that rejected into the water jacket, has appeared at the point of maximum temperature. Down the expansion curve the charge appears to lose heat to the walls at first rapidly, but toward the end the interchange of heat has nearly ceased.

'Taking the total heat of the gas used per explosion as 100, the indicated work is 37·4, heat rejected into the water jacket is 29, and that into the exhaust 33·6.

'The percentage (37·4) of useful work is the highest thermal efficiency which has to my knowledge been obtained in a test.

'The lower value has been taken for the calorific value of the gas, that is to say, the latent heat of the water formed during combustion has been taken from the total heat.'

The following table gives the particulars of the test and the analysis of the Manchester coal gas which was consumed:

TEST OF CROSSLEY GAS ENGINE WITH WATER INJECTION. (*Burstall*)

Cylinder 14 ins. diameter × 21 ins. stroke

Engine No. 48522.

Duration of test	6 hrs. 45 mins. (405 mins.)
Average revolutions per min.	166·02
Average explosions per min.	81·2
Average mean pressure	91·44 lbs. per sq. in.

Average indicated HP, total	60·5
Dead weight on brake	423 lbs.
Average pull on spring balance	9·5 lbs.
Effective diameter of brake wheel	7·61 ft.
Average brake HP	49·7
Mechanical efficiency	82·2 per cent.
Coal gas used per hour at 60° F. and 30 ins. bar.	712 cub. ft.
Gas per indicated HP per hour	11·77 cub. ft. at 60° F. and 30 ins. bar.
Gas per brake HP per hour	14·43 cub. ft. at 60° F. and 30 ins. bar.
Average calorific value of coal gas	578 B.Th.U. per cub. ft. at 60° F. and 30 ins. bar.
Thermal efficiency on indicated power	37·43 per cent.
Thermal efficiency on brake power	30·8 per cent.
Water discharged from jacket per minute	25·66 lbs.
Average rise in temperature of jacket water	77·52° F.
Mean temperature of exhaust as measured by Callendar pyrometer	381° C.
Water injected into cylinder per minute	0·131 lb.
Average percentage of carbon dioxide in exhaust gas	5·7 per cent.
Average percentage of oxygen in exhaust gas	9·6 per cent.

ANALYSIS OF MANCHESTER COAL GAS TAKEN AT MESSRS. CROSSLEY'S WORKS, OPENSHAW. (*Professor Percy Frankland*)

	Per cent.
Carbon dioxide (CO_2)	2·14
Oxygen (O_2)	0·00
*Heavy Hydrocarbons (C_nH_m)	6·8
Hydrogen (H_2)	42·9
Carbon monoxide (CO)	14·99
Marsh gas (CH_4)	30·93
Nitrogen (N_2)	2·24
	100·00

* This 6·8 per cent. of heavy hydrocarbons is equivalent to 8·5 per cent. of ethylene (C_2H_4).

Volumes of oxygen consumed by 100 volumes of gas	117·81
Heating value by calculation	582 B.Th.U. per cub. ft. at 60° F. and 30 ins. mercury

It is interesting to compare this test with Professor Hopkinson's, already described, as the compression is very much higher. Hopkinson's compression ratio $\frac{1}{r}$ was $\frac{1}{6\cdot37}$, giving an absolute compression pressure of 175 lbs. per sq. in., while Burstall's was $\frac{1}{8\cdot7}$, stated to give over 200 lbs. per sq. in. The compression pressure, however, is not definitely given for Burstall's test.

The air standard efficiency for $\frac{1}{6\cdot37}$ is 52·2 per cent., and for $\frac{1}{8\cdot7}$ it is 58·2 per cent., hence the efficiency ratios are $\frac{37}{52\cdot2} = 0\cdot71$ and $\frac{37\cdot4}{58\cdot2} = 0\cdot64$, so that even on the indicated efficiencies nothing has been gained by using the higher compression of over 200 lbs. per sq. in. Indeed, when brake results are compared, it is evident that efficiency has been lost by too high compression, as Hopkinson's best brake efficiency is 34·4 per cent., while Burstall only gets 30·8.

A test was made by Messrs. Crossley themselves on an engine, No. 49530, on January 13, 1905, with lower compression, of which they have published the following particulars:

TEST OF CROSSLEY GAS ENGINE WITH WATER INJECTION. (*Atkinson*)

Cylinder 11½ ins. diameter × 21 ins. stroke

Engine No. 49530.

Duration of test	6 hrs. 22 mins.
Mean revolutions per minute	185·17
Mean explosions per minute	92
Average mean pressure of diagrams	90·35
Average indicated horse-power	45·8
Net weight lifted by brake rope	488·7 lbs.
Effective circumference of brake rope	14·47 ft.
Average brake horse-power	39·68
Mechanical efficiency	86·64 per cent.
Coal gas used per hour corrected to a temperature of 60° F. and barometer pressure of 30 ins.	559·7 cub. ft.
Gas per indicated horse-power hour	12·22
Gas per brake horse-power hour	14·107
Average calorific value of gas as corrected to 60° F. and 30 ins. bar.	594 B.Th.U.'s
Thermal efficiency per indicated horse-power	35·35 per cent.
Thermal efficiency per brake horse-power	30·62 per cent.

Fig. 32 shows a diagram published with the test, from which it appears that the compression pressure is about 150 lbs. per sq. in. above atmosphere. and the explosion pressure 440 lbs. above atmosphere. It will be observed that the mechanical efficiency is higher than in the preceding test, 86·6 per cent. as against 82·2 per cent., so that although the indicated thermal efficiency is less, 35·3 per cent. against 37·4 per cent., yet the brake thermal efficiency is practically equal, 30·6 per cent. against 30·8 per cent.

The Crossley engines with cylinders of 11½ ins., 14 ins., and 11½ ins.

diameters tested respectively by Hopkinson, Burstall, and Atkinson, sufficiently illustrate the advance in economy made by this firm with engines of comparatively small dimensions.

The large gas engine movement received a great impetus by the demonstrations of the late Mr. B. H. Thwaite that blast furnace gas could be used in gas engines. Thwaite's system was put into operation at the Glasgow Iron Works in 1895, and shortly after that Continental and English makers attempted the construction of engines of greatly increased power.

In January 1901 Mr. Herbert A. Humphrey read a valuable paper before the Institution of Mechanical Engineers, entitled 'Power-Gas and Large Gas Engines for Central Stations,' in which he described the results he had obtained from very careful tests made with two large gas engines constructed respectively by Messrs. Crossley and the Premier Co., and used under his supervision by Messrs. Brunner, Mond & Co., Ltd., at their Winnington Works.

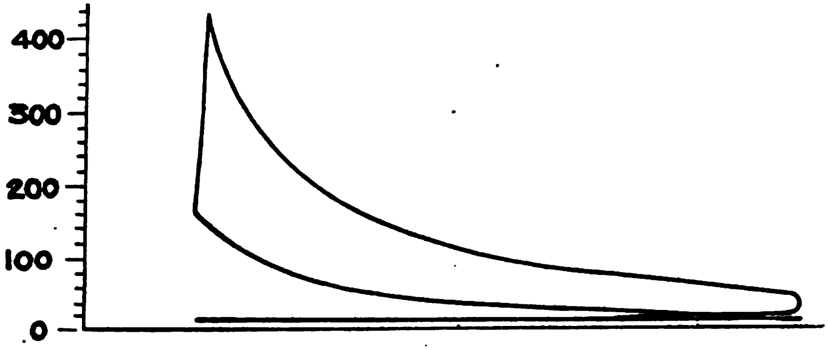


FIG. 32

The engines were supplied with gas made from bituminous fuel by Mond producers.

To enable us to compare early and late Crossley gas engines of considerable dimensions, it will be desirable to describe first the engine referred to by Mr. Humphrey in 1901, which had two cylinders, each of 26 ins. diameter, and having a stroke of 36 ins.; second, the engine tested by Dr. Nicolson in 1907, which also had two cylinders of 32 ins. diameter, the stroke being 26 ins.

The engine used by Mr. Humphrey shall be described first. It had two cylinders with open ends facing one another, and the connecting-rods worked on a crank common to both. The gas, air, and exhaust valves were all arranged to operate horizontally.

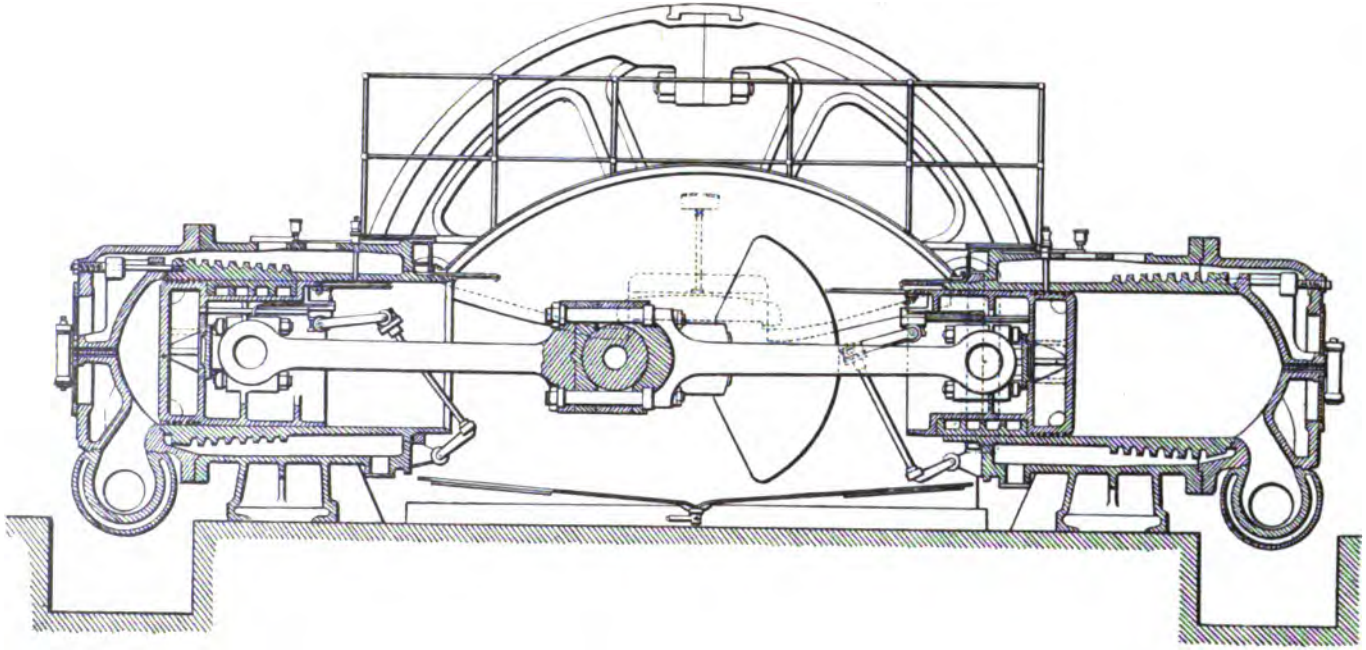


FIG. 33

Fig. 33 is a vertical longitudinal section of a Crossley engine of this type, but of larger dimensions than Mr. Humphrey's engine and differing in having watered pistons. In the engine shown the rated power is 600 BHP; the cylinders are 33 ins. diameter × 36 ins. stroke; it has watered pistons and exhaust valves; the exhaust valves are balanced; and in 1904 such engines were ignited either by hot tube or low-tension magneto. The exhaust valve cavity is shown at the lower end of the combustion space; it is of the piston balanced type, and is shown in longitudinal section on a larger scale at fig. 34. The exhaust valve seat is formed in a separate casting which is bolted on to the breech end from below. The upper passage leads from the cylinder to the valve; the lower is the exhaust passage leading to the exhaust piping. The valve is in one casting with a hollow cylinder which forms the valve stem; this cylinder is supplied with water by way of inlet and outlet pipes, and it has rings working in a removable sleeve, which make it pressure tight against the explosion. The diameter of the cylinder is somewhat less than that of the valve seat, so that its effect is to practically balance the valve still leaving a little pressure to hold it to its seat. It is pressed home by the spring shown, and the exhaust gases are discharged in the direction of the arrows. This valve is very effective, and the partial balance greatly reduces the stresses due to the opening of the valve against the terminal pressure of the exhaust. In large gas engines the opening effort is quite serious. The gas and charge inlet valves are of the ordinary Crossley horizontal type already described.

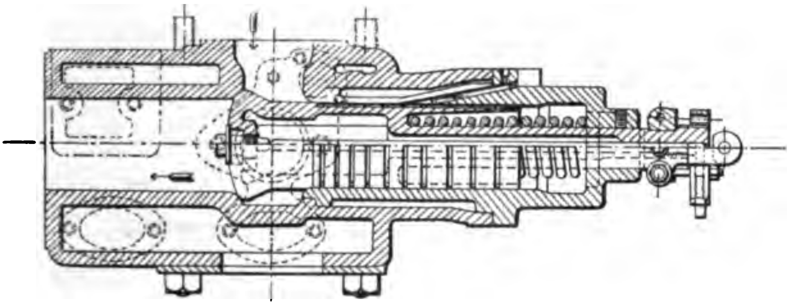


FIG. 34

Mr. Humphrey gives the following particulars of two tests made with the 400 HP engine on April 10 and July 2, 1900.

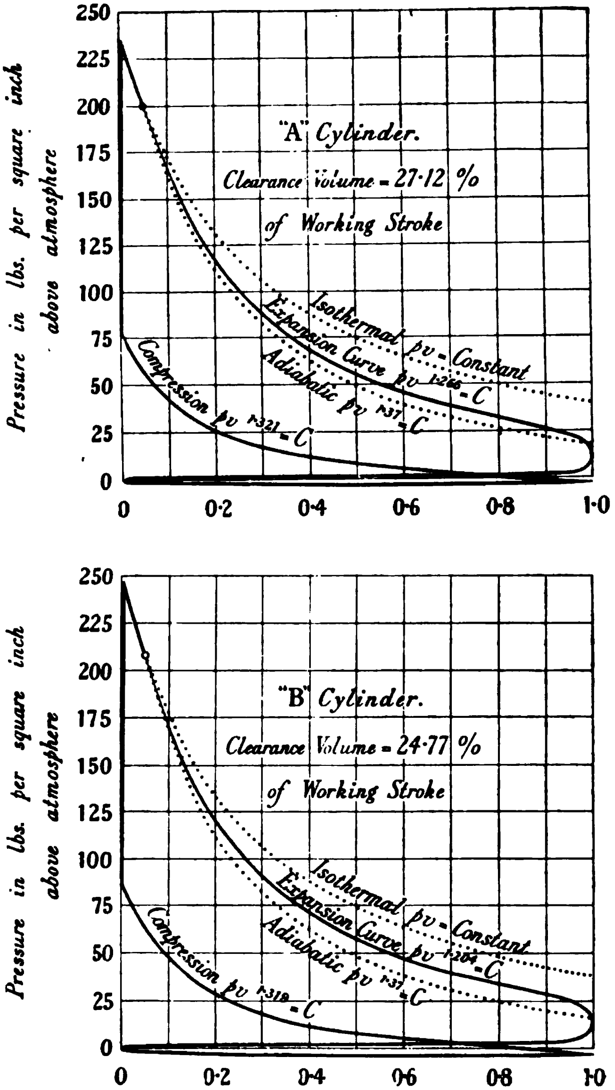

Pressure in lbs. per square inch above atmosphere
250
225
200
175
150
125
100
75
50
25
0
0
0·2
0·4
0·6
0·8
1·0
"A" Cylinder.
Clearance Volume = 27·12 %
of Working Stroke
Isothermal pv = Constant
Expansion Curve pv 1.266 = C
Adiabatic pv 1.37 = C
Compression pv 1.321 = C
Pressure in lbs. per square inch above atmosphere
250
225
200
175
150
125
100
75
50
25
0
0
0·2
0·4
0·6
0·8
1·0
"B" Cylinder.
Clearance Volume = 24·77 %
of Working Stroke
Isothermal pv = Constant
Expansion Curve pv 1.204 = C
Adiabatic pv 1.37 = C
Compression pv 1.318 = C

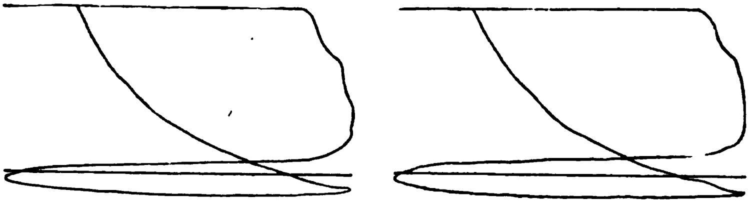
FIG. 35

TRIAL OF 400 HP CROSSLEY GAS ENGINE CONSUMING MOND GAS. (*Humphrey*)

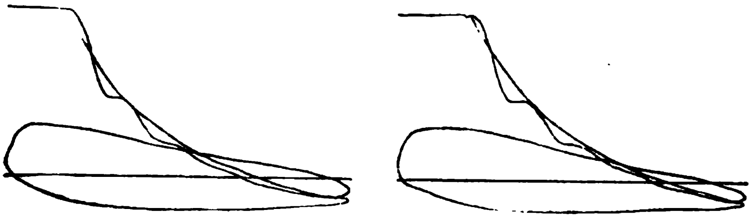
Engine, diameter of two cylinders	26 ins.
„ length of stroke	36 ins.
Speed, revolutions per minute	150
Clearance volume, A cylinder	2·996 cub. ft.
„ „ B „	2·740 cub. ft.
Clearance volume as per cent. of working stroke:	
A cylinder	24·12 per cent.
B „	24·77 per cent.
Clearance volume as per cent. of total volume:	
A cylinder	21·34 per cent.
B „	19·85 per cent.

$\frac{1}{r}$ for A = $\frac{1}{4{\cdot}59}$, for B = $\frac{1}{5{\cdot}3}$

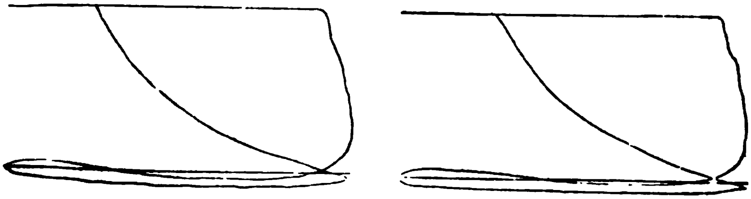
A CYLINDER B CYLINDER,



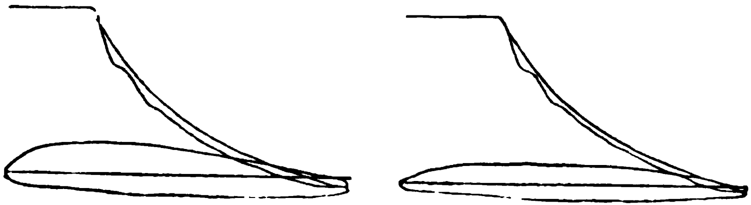
150 revs. per minute; firing cycle.



150 revs. per minute; taking air only.



94 revs. per minute; firing cycle.



94 revs. per minute; taking air only.

FIG. 36

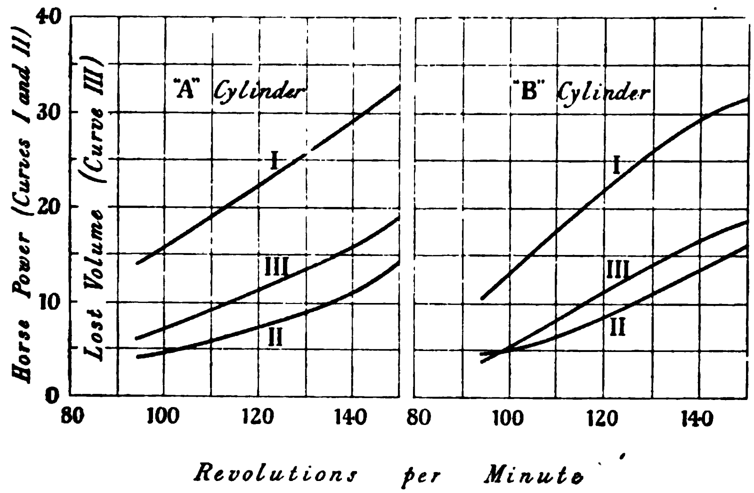
Both cylinders taking gas every cycle. No misses	Trial at normal working load	Trial at full load
Date of trial	July 2, 1900	April 10, 1900
Duration of trial	2 hours	6 hours
Average output of dynamo, amperes	2208·0	2268·4
" " " volts	98·6	110·0
" " " EHP	291·8	334·4
Average revolutions per minute	152·4	148·5
Mean effective pressure, average for 'A' and 'B' cylinders	51·37	60·43
Average indicated horse-power	377·9	432·9
Total Mond gas as measured by meter, cubic feet	49,640	176,820
Gas used per hour, reduced to 0° C. and 760 mm.	22,711	27,162
Analysis of dry gas:		
CO_2 vol. per cent.	14·5	15·0
CO " " "	12·0	11·5
H " " "	29·0	28·5
CH_4 " " "	2·0	2·1
N " " "	42·5	42·9
Efficiency of dynamo, maker's figure, per cent.	93	93
Brake horse-power	313·8	359·6
IHP absorbed in fluid resistances	29·20	27·80
" " " engine friction	34·90	45·53
Total frictional losses, HP	64·10	73·33
Mechanical efficiency of engine:		
Excluding fluid losses	90·76	89·48
Including fluid losses	83·04	83·06
Combined efficiency, EHP/IHP, per cent.	77·23	77·25
Mond gas used (cubic feet at 0° C.):		
Per IHP hour	60·09	62·74
" BHP "	72·37	75·53
" EHP "	77·82	81·22

	Higher	Lower	Higher	Lower
Calorific value of gas (at 0° C.):				
Kilo-calories per cubic metre	1444·5	1287·9	1423·5	1268·3
B.Th.U. per cubic foot	162·2	144·6	159·9	142·5
Thermal efficiency, per cent:				
Calculated on the IHP	26·23	29·42	25·49	28·61
" " BHP	21·78	24·43	21·17	23·76
" " EHP	20·26	22·73	19·69	22·09

NOTE.—Between the two tests recorded above, the pistons were slightly reduced in diameter, giving greater freedom of working for the normal load trial; the exhaust pipes were also altered, so as to remove them to the outside of the building. More gas per stroke was used at full load, the quantity corresponding with greatest economy being exceeded.

Fig. 35 shows diagrams from A and B cylinders representing the average of all the diagrams taken during the official trial of April 10, 1900. The average of the mean pressures from A and B is 60·43 lbs. per sq. in. The pressure of compression in B is about 85 lbs. per sq. in. above atmosphere, while that in A is just over 75 lbs., while the pressure of explosion is about 245 lbs. above atmosphere in B and about 235 lbs. in A.

Fig. 36 shows charging diagrams taken at 150 and 94 revolutions per minute for firing cycle and governing taking air only. In this engine governing was accomplished by means of two cams and missing. The cams served to vary the mean pressure down to 78 per cent. of the maximum, and for further governing charges were cut out. The proportion of Mond gas to air, like producer gas, is very great, about equal charges of Mond gas and air, and the throttle caused by stopping the gas flow is very considerable, as in this early engine no provision was made for increasing the air passage area when gas was cut off. The charging losses were thus much lower when firing than when running light.



I.—Horse-power lost when cylinder is run idle, taking in and discharging air only.
II.—Horse-power lost in fluid resistance when exploding at every cycle.
III.—Percentage of return stroke performed when compression curve crosses the atmospheric line in ordinary working cycle.

NOTE.—All three curves were obtained from 'bottom loop' diagrams taken with a light spring.

FIG. 37

Fig. 37 gives interesting curves calculated from fig. 36, showing horse-power lost in fluid resistance at different speeds, firing, and taking air only. The curves show that at 150 revolutions per minute the power lost in fluid resistance when exploding at every cycle is about 15 HP, while over 30 HP is so lost when the engine is running idle, taking in and discharging air only.

Interesting experiments were made with varying Mond gas charges per stroke, keeping the engine running at 120 revolutions per minute. The results are shown by curves at fig. 38, from which it appears that the maximum economy of gas is obtained with a charge of 2·7 cub. ft. of Mond gas per cylinder, while the maximum power requires a charge of 3 cub. ft. Above 3 cub. ft. the power suddenly falls off, and the engine misses fire. The weakest charge which will run is about 2·2 cub. ft. Between 2·3 and 3 cub. ft. the indicated horse-power obtained ranges from 110 to 190 in cylinder B and 100 to 163 in cylinder A.

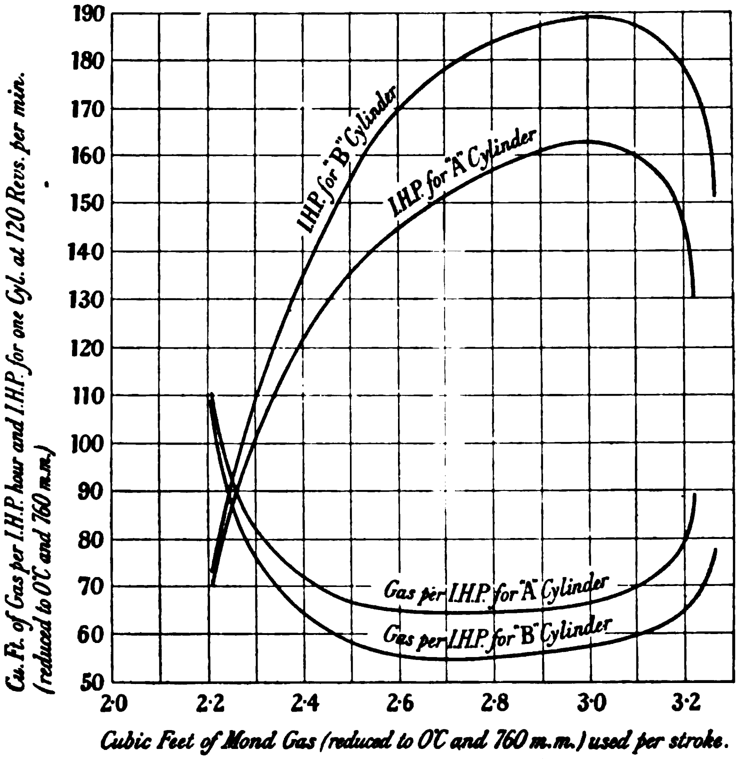
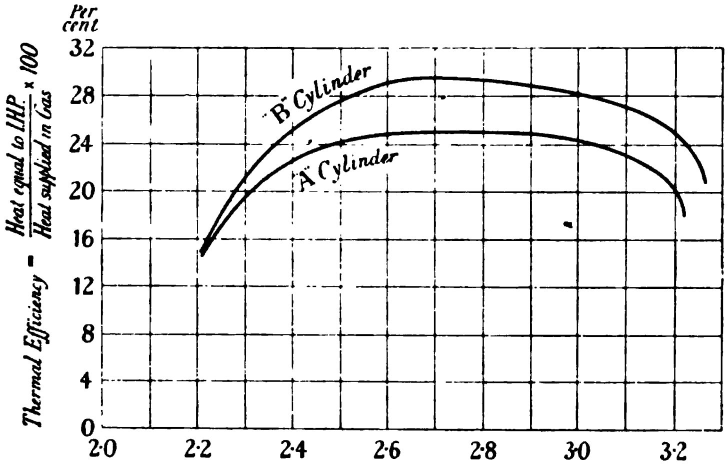


FIG. 38

Fig. 39 shows the variation in indicated thermal efficiency with variation of gas charge volume; here a maximum efficiency of over 29 per cent. is obtained by the B cylinder with a charge of 2·7 cub. ft. of gas, while only 25 per cent. is given by the A cylinder for the same charge.

The experiments proved that this engine could be governed from full to nearly half load without cutting out ignitions, and down to two-thirds indicated load the efficiency does not fall off very much.

Mr. Humphrey states that this 400 HP engine is fitted with a graduated or stepped gas die, upon which the gas lever and knife edge strike. The governor determines the position of this die, and consequently the amount of gas admitted. Mr. Humphrey's test is of great importance, as all the gas used was measured by a large gas meter. It is rare to find tests of actual gas measurements made with engines of a cylinder diameter so large as 26 ins. and stroke 36 ins.; the undertaking is a formidable one, as may be seen from the fact that a large station wet meter of 50,000 cub. ft. per hour capacity was erected especially for the experiments.



Cubic Feet of Mond Gas (reduced to 0°C. and 760 m.m.) used per stroke.

FIG. 39

The engine is also interesting as one of the largest engines at the date which has run steadily almost night and day for months at a time without requiring the use of watered pistons. In the discussion which followed the paper Mr. Humphrey stated that the engine had improved during its six months' heavy load running, so that the thermal efficiency on the IHP had increased from 26·2 to 27·2. This is equal to 30·5 per cent. IHP thermal efficiency on the lower heating value, an excellent result for an engine without water in its piston working at the low compression used.

Dr. Nicolson's test of a large Crossley engine was made at Manchester in 1907.

Fig. 40 is a vertical longitudinal section of a similar engine, which consists of two single-acting, open-ended cylinders in tandem, driving one crank by a connecting-rod. The cylinders were 32 ins. diameter, and the stroke was 26 ins.

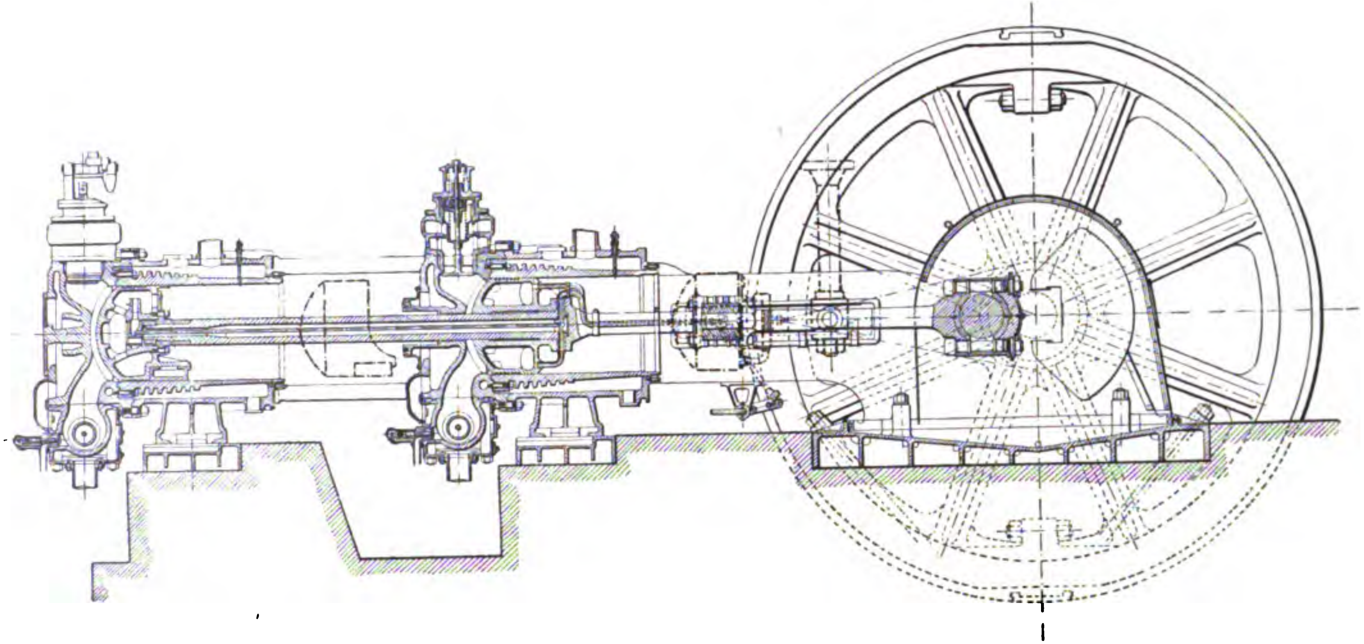


FIG. 40

The small end of the connecting-rod is coupled to a crosshead which runs in double guides ; the connecting-rod end is forked, and so leaves a space between the forks on the crosshead pin. In this space there is fitted a vertically swivelling distance piece, which connects between the crosshead pin and a faced and spigoted end on the first piston. The flange end of the distance piece is bolted to the faced surface of the piston, and so are pipe connections for water supply. When the fastening bolts and the pipe connecting bolts are removed, the distance piece can be pulled forward, and when clear of the piston swung up vertically on the crosshead pin, as shown in dotted lines. The second piston is connected to the first by a hollow piston rod, passing through a packing gland in the first cylinder cover. When the front distance piece is swung up vertically, the two pistons can be drawn forward out of their cylinders to the positions shown in dotted lines. By this simple device the pistons can be examined without removing them from their rod, and ready access can be obtained to the cylinders, which can be thoroughly cleaned or examined as well as the pistons. Messrs. Crossley consider this a very important advantage over the double-acting cylinders with cylinder covers ; they consider that it is desirable to be able to see the condition of the cylinders while the engine is running ; the exact amount of lubrication required may be readily judged, and it can be seen whether the pistons are tight or not. Messrs. Crossley consider that single-acting pistons are more reliable than double-acting. This type, too, requires only one piston rod gland, as compared with four commonly used with tandem double-acting engines, which shall be described later on.

The single-acting, open-ended cylinders also allow the use of the types of liner and breech ends which act so favourably in the smaller engines ; that is, they allow of a liner fixed at the breech end but free to expand at the open end so as to allow for the difference in temperature between liner and water jacket casing.

The water for cooling the pistons passes in at one side of the crosshead pin, flows by a pipe connection lying along one side of the distance piece into a passage in the first piston, and along the hollow rod to the second piston, whence it returns by a central tube to the first piston, and discharges into another pipe connection on the other side of the distance piece, and flows away at the other side of the crosshead pin. The pivoted pipe system supplies water under pressure to the crosshead pin at one side, and a similar system carries it away at the other side.

The exhaust valves are of the piston balanced type described at p. 45. The gas valves act by varying the point of admission of gas to increase or diminish the quantity present in the charge by the

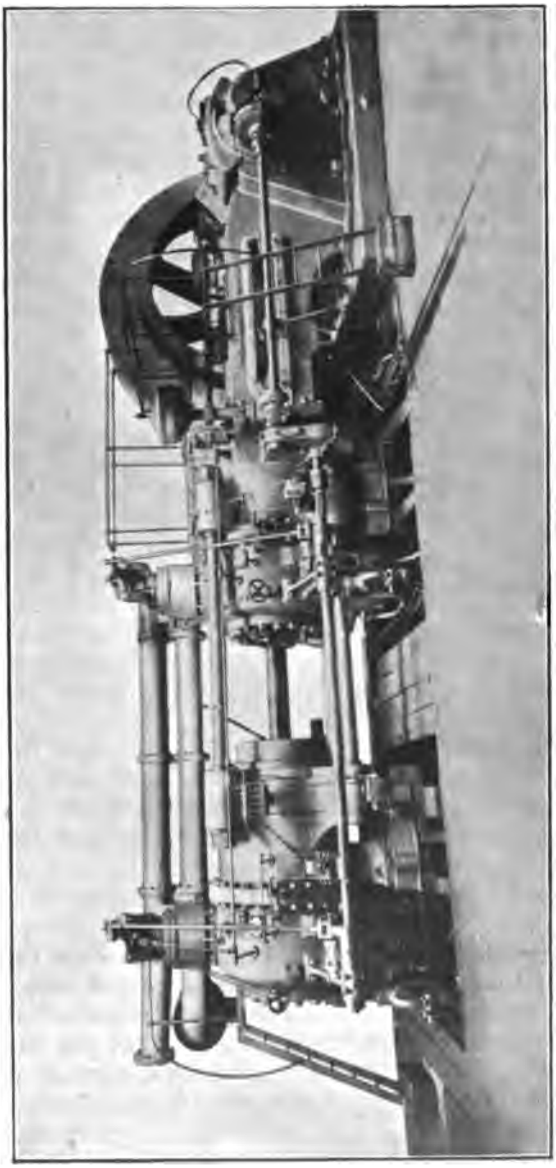


FIG. 41

method known as 'quality governing' to be described later. Here it may be stated that the gas valve is controlled by a very ingenious device of pneumatic piston, whose action is determined by the movement of a small balanced plug valve actuated by the governor. This governing device has proved itself quite reliable, and is free from sticking, even when the gas supplied to the engine contains a little tar. The compressed valves for supplying air to the cylinders to enable the engine to be started are also indicated.

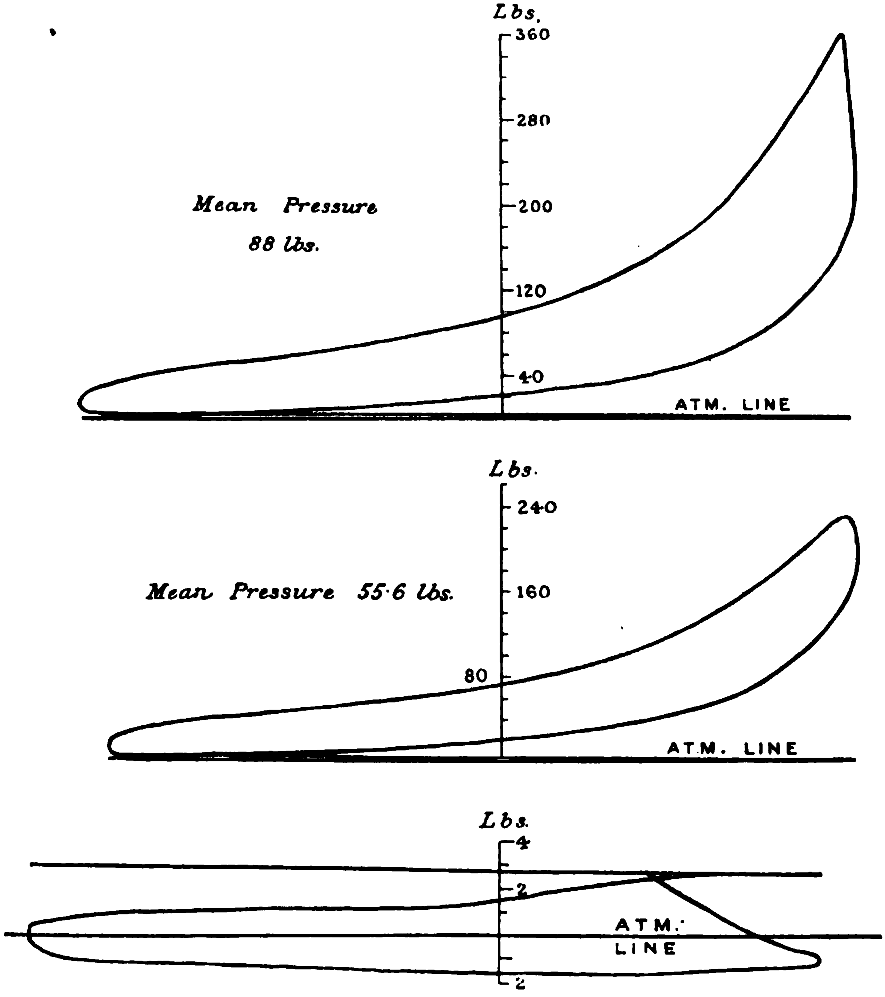


FIG. 42

The engine is fitted with a system of forced lubrication which supplies oil to the cylinders, the balancing pistons of the exhaust valves, and the piston rod gland. A distributing box is applied from which separate branches pass to each part requiring lubrication; each separate delivery contains an oil regulating valve, from which oil passes through the tube of a sight feed.

A photograph of an engine entirely similar in construction but of larger dimensions is shown at fig. 41; in it the cylinders are 38 ins. diameter, and the stroke 39 ins. Many of the external details of the engine are very clearly seen; the way-shaft receiving motion from the crankshaft is geared to it by bevel wheels to run at the same speed, and the valve shaft is geared to this rapid running first shaft by spur wheels in the usual two to one ratio. The first way-shaft drives the governor, which actuates a long rocking shaft connecting by arms and links to the pneumatic relay valves operating the pneumatically controlled gas valve. Ignition is accomplished by four magnetos giving two igniting contacts to each breech end; these magnetos are tripped from the valve shaft. Two hand wheels are seen which serve to adjust the air throttle. Gas and air are separately fed to the two charging valves by the two pipes shown. The high pressure air starting valves can be seen attached to the exhaust valve casings, and the pipes controlled by cam operated valves leading to the starting valves may also be seen.

The results of Dr. Nicholson's test are as follows:

TEST OF CROSSLEY SINGLE-ACTING TANDEM CYLINDER GAS ENGINE. (*Nicolson*)

Two Cylinders: diameter 32 ins.; stroke 26 ins.

Rated revolutions per minute; explosions consecutive		120
Brake horse-power; engine loaded by friction brake		559
Bituminous fuel gas used per BHP hour (at 0° C. and 760 mm. mercury)		51·94 cub. ft.
Heat supplied per BHP hour		8128 B.Th.U.
Thermal efficiency on brake power		31·32 per cent.
Analysis of Gas (Nicolson):		
Carbon dioxide (CO_2)	11·4	
Unsaturated hydrocarbons	None	
Oxygen (O_2)	1·0	
Carbon monoxide (CO)	15·1	
Hydrogen (H_2)	24·3	
Methane (CH_4)	3·5	
Nitrogen (difference)	44·7	
	100·0	
Lower calorific value calculated from analysis per cubic foot at 0° C. and 760 mm. pressure		156·5 B.Th.U.
Lower calorific value by Junker calorimeter per cubic foot at atmospheric temperature and pressure		149 B.Th.U.

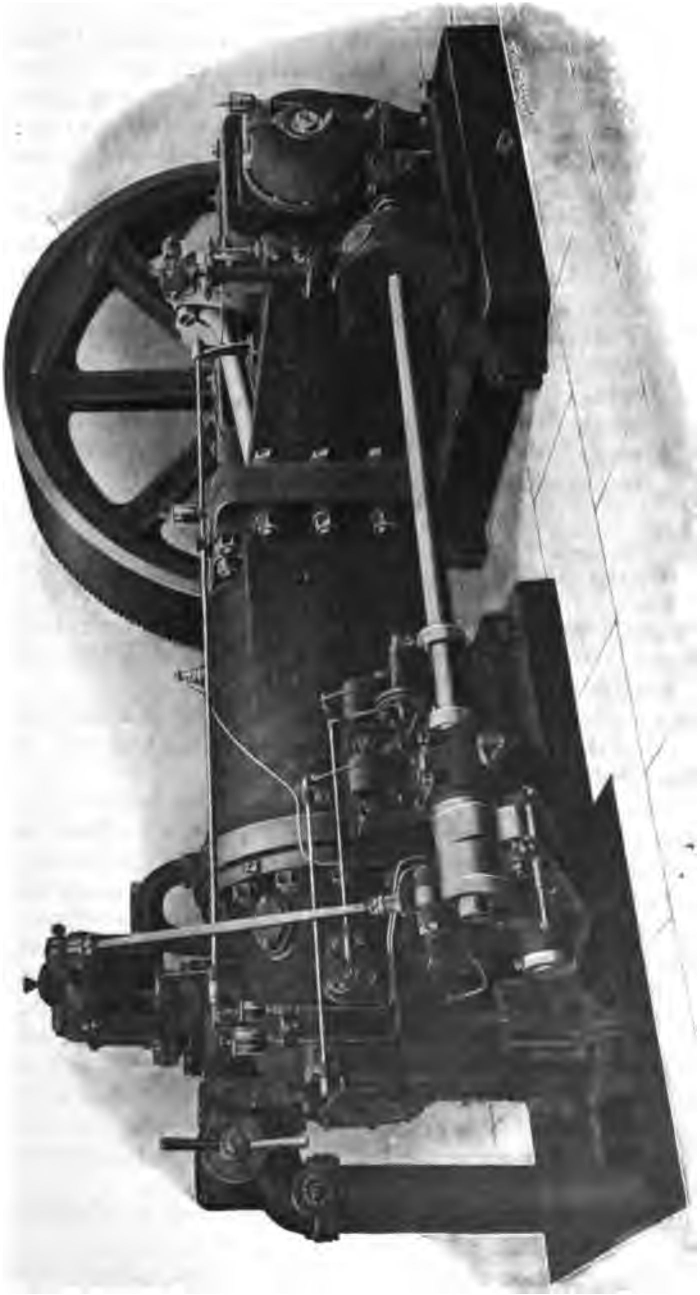


FIG. 43

Dr. Nicolson did not give the IHP of the engine, as he preferred to give thermal efficiency on the more accurately obtained brake power, but Mr. Atkinson states that the mechanical efficiency of the engine is 88 per cent.; so that the IHP corresponding to 559 BHP is 613·6; the engine thus gave one indicated horse-power for one hour on 7153 B.Th.U., showing a thermal efficiency on the indicated horse-power of 35·6 per cent.

Mr. Atkinson considers that 31·32 per cent. is the highest indicated efficiency that had been obtained up to April 1908 in any well authenticated trial with a large gas engine; higher efficiencies had been obtained with smaller engines.

Mr. Atkinson gives indicator diagrams taken from this engine while governing from full and half load.

Fig. 42 shows these diagrams.

It will be seen that the mean pressure at full load is 88 lbs. per sq. in., and at half load about 55·6 lbs. A light spring diagram is given showing the charging and discharging resistances.

CROSSLEY OTTO HORIZONTAL ENGINE. 175 BHP. (1910)

Fig. 43 shows a general view of a new single cylinder horizontal engine produced by Messrs. Crossley in 1910 and exhibited by them at the Brussels Exhibition.

Fig. 44 is a vertical longitudinal section, and fig. 45 is a vertical transverse section through the inlet and exhaust valves and part of the combustion chamber.

From the illustrations it will be seen that the engine bed is of the built up girder type, and the rear portion of the girder is designed to form the water jacket casing and carry the whole length of the cylinder, so that no part overhangs the end of the bed. The combustion chamber A is of conoidal form, and it is bolted to the end of the cylinder B and so arranged that water connection through the flange and joint is quite avoided. Water is supplied to the separate cylinder and combustion chamber jackets C and D by the forked supply pipe D_1, and is carried away by the similarly forked pipe E. The charge inlet valve F and the exhaust valve F are arranged vertically at the end of the conoidal space. The exhaust valve is watered and its hollow stem moves in a bushed guide. The gas valve G is of the pneumatic type described at p. 55, and it varies the cut off point of the gas supply by the operation of the governor on the air control plug H.

G_1 is the gas supply passage with its regulating cock G_2, and H_1 is the air supply pipe with its control valve H_2.

Two low-tension ignition plugs are used, one of which is shown at I (fig. 45); the other is applied at the end of the cylinder and acts through the aperture I_1 (fig. 44).

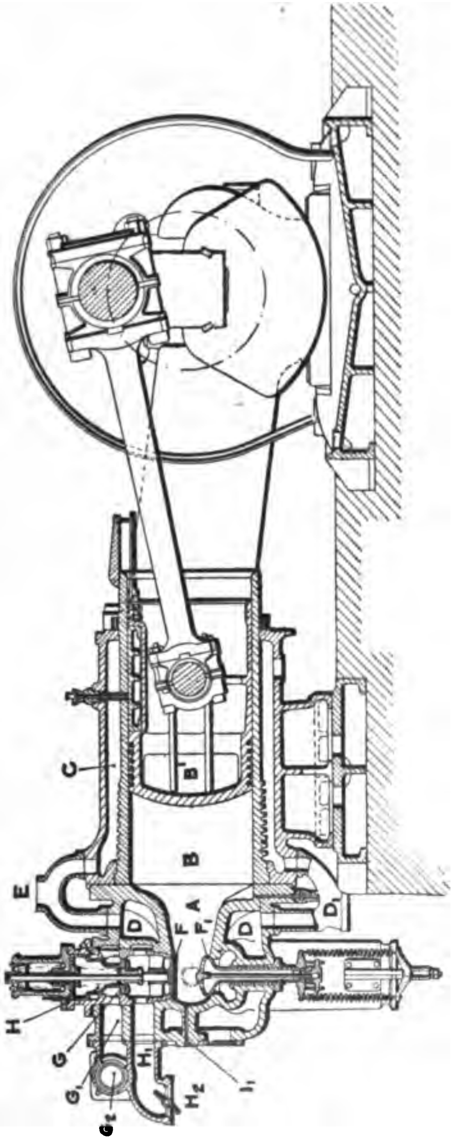


FIG. 44

The air pressure starting valve is shown at J (fig. 45).

The exhaust pipe K is water jacketed for the first part of its length.

The piston B′ depends on air cooling.

The cylinder and exhaust valve spindle are lubricated by force pump, and ring lubrication is applied to the main bearings, outer bearings, and side shaft bearings.

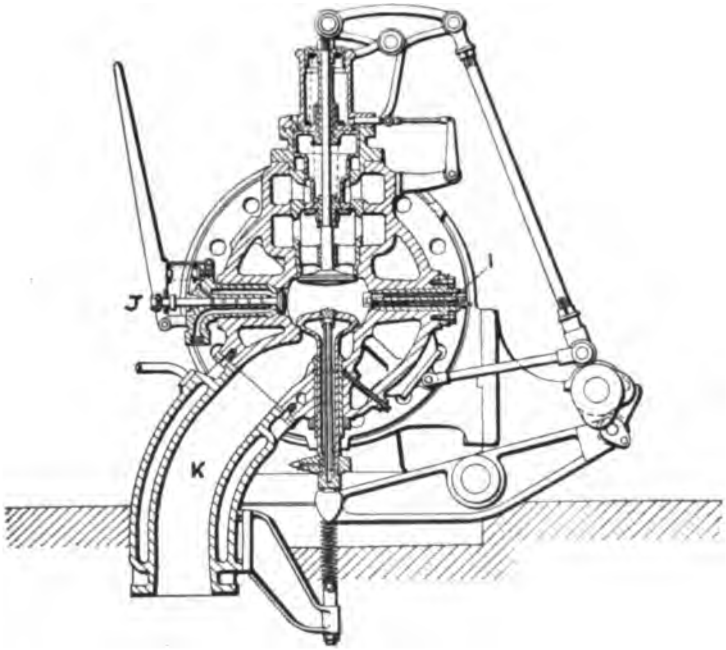


FIG. 45

Two magnetos are used, as can be readily seen from fig. 43, and a variable timing device is fitted which allows the ignition to be advanced or retarded while the engine is running.

CROSSLEY OTTO VERTICAL ENGINE. 250 BHP. (1910)

Fig. 46 is a vertical section through one of the cylinders of a four-cylinder vertical engine built by Messrs. Crossley in 1910. The cylinder diameters are each 16 ins. and the stroke 18 ins. Rated speed, 250 revolutions per minute.

The cylinder and the water jacket casing are cast in one piece, but the casing is separated from the cylinder at the lower end, and a flexible joint is made there to contain the water; by this device the casing allows the cylinder to freely expand in order to avoid dangerous stresses due to the difference of expansion between the hotter cylinder and the casing.

The cylinder heads or covers are hollow cylindrical castings having a conical ground-in seat making a metal-to-metal joint within the

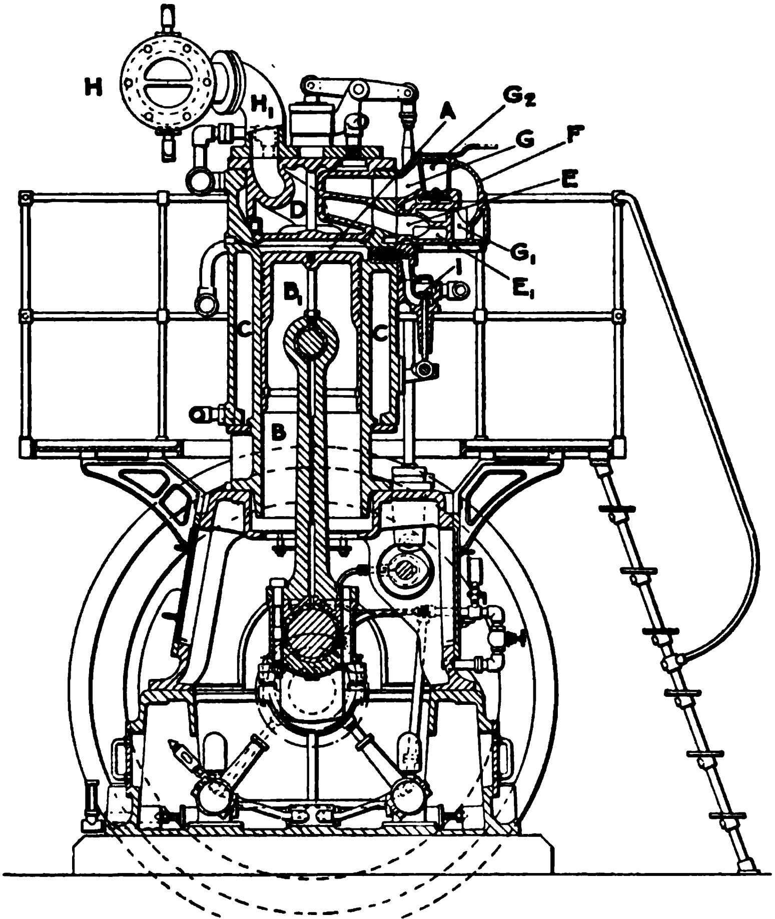


Fig. 46

upper end of the cylinder. The combustion space is formed by the clearance between the piston and inner surface of the cover. The inlet and exhaust valves fit into the cylinder heads, and so open directly into the combustion space without port or passage. This arrangement leaves the combustion chamber free from ports or pockets ; it also allows of a symmetrical and simple cylinder casting. The valve seats are formed in the heads or covers, and so can be got at readily. In this engine gas and air are supplied separately to the charge inlet valves, and so, if a back ignition does occur, it affects one cylinder only, as there is no explosive mixture stored in the supply pipes. A is the combustion chamber; B the cylinder; B_1 the piston; C the cylinder water jacket; D the head or cover water jacket. Each head or cover carries an air port, E, and a gas port, G, which lead to the inlet valve. A trunk or manifold, F, passes along the front of the cylinders, and bolts to each; it carries passages or ports E_1 for air, and G_1 for gas. The gas can be cut off at each cylinder by the gas cock G_2. The exhaust pipes H_1 likewise connect to an exhaust manifold, H, which runs from cylinder to cylinder; this manifold is water jacketed. Any of the cylinder heads can be separately removed without dismantling either inlet or exhaust manifolds. The short exhaust pipe H_1 is unbolted, the valve covers are disconnected, and the head or cover can then be lifted out after removing the holding down nuts.

By this arrangement the gas and air main is on the opposite side of the engine to the exhaust main, and so can be kept cool more easily.

High-tension ignition plugs are used, and governing accomplished by throttling the gas and air supplies independently by separate gas and air valves mounted on the same spindle.

Starting is accomplished by compressed air. I is the starting valve.

The engine is lubricated by forced circulation of oil; two oil pumps for this purpose are shown in the base; they are driven from one eccentric, and the oscillating movement produces the necessary valve action.

THE NATIONAL GAS ENGINE COMPANY'S OTTO CYCLE ENGINES

The thermal efficiency of the engines built by this company is shown by the table on p. 60, which summarises the results of careful tests made by Professor Robinson in 1898, the Institution of Civil Engineers' Committee on the Standards of Efficiency of Internal Combustion Engines in 1905, Mr. Brydges of London in 1909, and Dugald Clerk and Mr. Stead of the National Co. in 1911.

In the standard National engines no device is adopted to keep down pre-ignition, and the compressions used are safe compressions such as can be depended upon without injecting water spray. Accordingly the compression ratios of the National engines are as a maximum for horizontal engines about $\frac{1}{r} = \frac{1}{5 \cdot 5}$, corresponding to compressions about 125 lbs. per sq. in. above atmosphere, and for tandem vertical engines $\frac{1}{r} = \frac{1}{6}$, about 160 lbs. per sq. in. above atmosphere. By adopting water spray injection higher compressions may be used, as is shown in the Crossley engine tests on p. 30, where the compressions in Atkinson's and Hopkinson's experiments are respectively 150 and 160 lbs. above atmosphere, and the brake thermal efficiencies 30·6 per cent. and 32 per cent. Hopkinson's test was too short in duration to be considered as representing the average performance of the engine, although Atkinson's very fairly does this by a full load run of over six hours. Burstall's test, although conducted on a larger engine and with a compression over 200 lbs. per sq. in., gave only 30·8 per cent., also for a six hours' run.

INDICATED AND BRAKE THERMAL EFFICIENCY OF NATIONAL-OTTO ENGINES. 1898–1911

Names of experimenters	Year	Dimensions of engine		Indicated thermal efficiency	Brake thermal efficiency	Mechanical efficiency	Compression, lbs. per sq. in. above atmosphere
		Dia.	Str.	Per cent.	Per cent.	Per cent.	
Robinson .	1898	10"	× 18"	28·7	25·0	87·0	—
Inst.C.E. .	1905	14"	× 22"	35·0	29·9	85·5	125
,, .	1905	9"	× 17"	33·3	28·3	85·0	125
,, .	1905	5·5"	× 10"	31·8	26·7	83·8	125
*Brydges .	1909	22½"	× 30"	33·0	28·1	85·0	—
Clerk .	1911	9"	× 17"	33·4	28·4	85·0	125
Stead .	1911	23"	× 24"	30·0	27·0	90·0	160

* Engine and suction producer plant working with Scotch anthracite. Consumption 0·845 lb. per BHP hour; thermal value taken at 13,500 B.Th.U. per lb. Efficiency of producer taken as 80 per cent., and mechanical efficiency of engine as 85 per cent.

The National engine of similar dimensions to that used in Burstall's test gave a brake efficiency of 29·9 per cent., a result which justifies the engineers of the company in keeping down the compression to 125 lbs. It may be taken that any good engine of about 14 ins. diameter cylinder × 22 ins. stroke can give a brake thermal efficiency of 30 per cent. with a value of $\frac{1}{r} = \frac{1}{5 \cdot 5}$, and one of the most interesting results of the Institution of Civil Engineers' tests shows that over 28 per cent. may be obtained from a 9 ins. × 17 ins. engine, and over 26 per cent. from an engine so small as 5·5 ins. × 10 ins.

The National Company now build gas engines of from ¾ to 1500 BHP. Up to 10 BHP the engines are of the overhung cylinder type, as illustrated at figs. 92, 93, and 94 of the first volume; above that power the girder bed with supported cylinder is adopted. Horizontal single cylinder engines are made up to 185 BHP; above that power multiple cylinders are adopted, both horizontal and vertical.

To illustrate the details of the latest types of horizontal engines three sizes have been selected for discussion, of 14 ins., 16 ins., and 22 ins. diameter cylinders respectively. The following table gives particulars as to mean effective pressures and power obtained, using coal gas, producer gas from anthracite, producer gas from coke, and benzene as fuel:

POWER OF NATIONAL GAS ENGINES WITH COAL GAS, PRODUCER GASES, AND BENZENE

—		Mean pressure	Max. IHP	Max. BHP	Rated BHP	BHP 10 % overload
Dia. Stroke. 14" x 21" at 230 revs. per minute	Coal gas	95	89·0	75·5	68·0	74·8
	Anthracite producer gas	80	75·0	61·5	55·0	60·5
	Coke producer gas	75	70·3	56·8	50·0	55·0
	Benzene	85	79·6	66·1	60·0	66·0
Dia. Stroke. 16" x 22" at 210 revs. per minute	Coal gas	95	111·2	94·2	85·0	93·5
	Anthracite producer gas	80	93·7	76·7	70·0	77·0
	Coke producer gas	75	88·0	71·0	65·0	71·5
	Benzene	85	99·5	82·5	75·0	82·5
Dia. Stroke. 22" x 34" at 160 revs. per minute	Coal gas	95	248·0	210·5	185·0	203·5
	Anthracite producer gas	80	208·5	171·0	150·0	165·0
	Coke producer gas	75	195·5	158·0	140·0	154·0
	Benzene	85	222·0	184·5	160·0	176·0

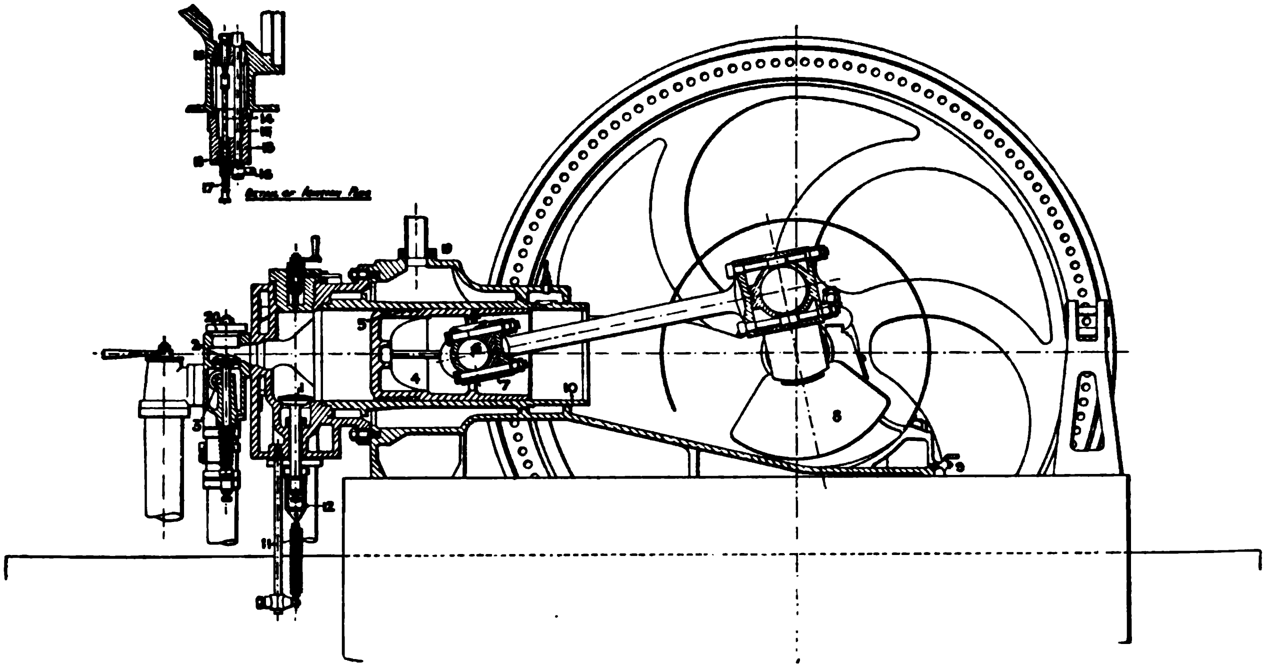
Fig. 47 is a vertical longitudinal section of the standard single-cylinder horizontal engine of 14 ins. cylinder diameter and 21 ins. stroke as produced by the National Gas Engine Company.

A horizontal section of the ignition plug is also shown on a larger scale.

Fig. 48 is an external longitudinal elevation, and fig. 49 an external end elevation.

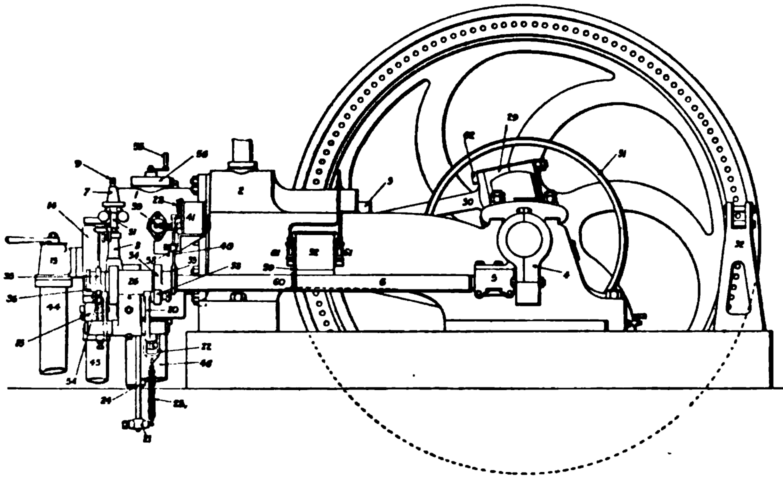
Lists of the parts are given under these figures, so that but little explanation is required.

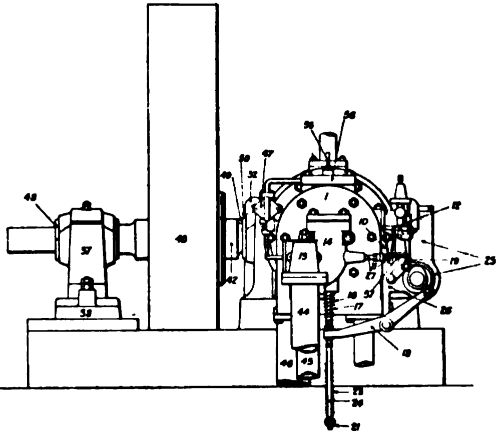
It will be seen that the engine bed is of the girder type, and the two side girders are connected together at the rear end by the water jacket casing which forms part of the bed; the girders carry the main bearings, and are connected at their front ends by a cross girder and joined by the lower part of the casting back to the casing to form



1. Exhaust valve
2. Air admission valve
3. Gas admission valve
4. Piston
5. Piston rings
6. Piston pin
7. Piston pin brasses
8. Crank balance weights
9. Oil drain cock
10. Cylinder liner
11. Exhaust spring holder
12. Exhaust spring shackle
13. Ignition plug (electric)
14. Fast spindle
15. Loose spindle
16. Tappet lever
17. Brass terminal
18. Porcelain tubes
19. Loose flange for water pipe
20. Air valve cover

FIG. 47





1. Cylinder casing
2. Bed
3. Cylinder liner
4. Crank worm wheel case
5. Bed S.S. bracket
6. Side shaft
7. Governor weight
8. Governor pillar
9. Governor speed adjusting nuts
10. Governor hanging link
11. Governor die
12. Governor lever
13. Gas cock
14. Air and gas valve casing
15. Air throttle (for suction engines only)
16. Admission valve spindle (air)
17. Admission spring (air)
18. Admission lever (air)
19. Gas lever
20. Exhaust lever
21. Exhaust spring holder
22. Exhaust spring shackle
23. Exhaust spring
24. Exhaust spring holder stud
25. S.S. gear wheels
26. Cylinder S.S. bracket
27. Gas valve spindle
28. Magneto spring
29. Crank pin brasses
30. Con. rod
31. Oil splash guard
32. Barring bracket
33. Ignition ecc. sheave and strap
34. Exhaust cam
35. Admission cam (air)
36. Gas cam
37. Outer pedestal
38. Outer pedestal sole plate
39. Electric ignition plug
40. Trip rod
41. Magneto
42. Crankshaft
43. Flywheel
44. Gas pipe
45. Air pipe
46. Exhaust pipe
47. Hand pump for starting engine
48. Outer pedestal brasses
49. Engine bearing brasses, flywheel side
50. Engine bearing brasses, S.S. side
51. Governor sleeve
52. Forced lubricator pump
53. Exhaust roller
54. Admission roller (air)
55. Handle for self starter
56. Exhaust valve cover
57. Gas lever knife
58. Regulating handle for electric ignition
59. Oil pump lever
60. Oil pump cam
61. Oil pump sight feed
62. Con. rod bolts, crank end

FIG. 48

FIG. 49

an oil collecting trough. The bed casting forms pedestals at back and front, as shown. The cylinder liner is carried by and bolted to the combustion chamber casting, which is bolted to the faced end of the water jacket casing. The front end of the liner is jointed to the casing by a rubber packing, so that it is free to expand relatively to the casing. The combustion chamber is water jacketed, and carries the exhaust valve seat and discharge passage connecting by flange to the exhaust pipe ; it also carries the exhaust valve cover, which serves to contain a starting valve to enable the engine to be started by means of an inflammable charge pumped into the cylinder by a hand pump seen at fig. 49. The water inlet pipe connects at the right side of the combustion chamber jacket (see fig. 49), and the water after passing from the combustion chamber jacket to the main water jacket leaves the engine by the pipe shown (fig. 47). The exhaust valve spindle is held in a bushed guide.

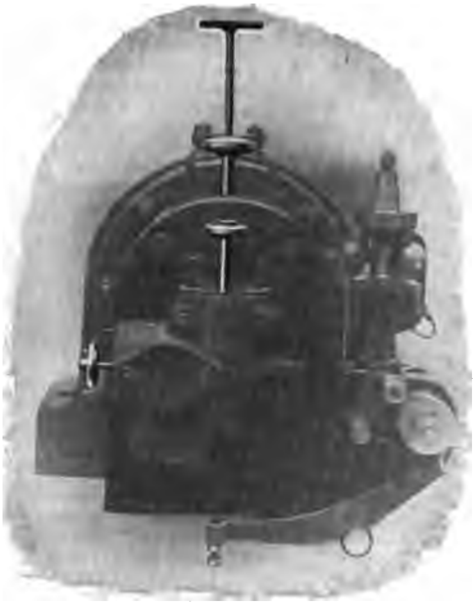


FIG. 50

The charge inlet and gas valves are contained in a separate casing which for this size of engine is not water jacketed. This casing carries gas and air connections and is bolted to the combustion chamber, and it also carries the inlet and gas valve covers, as shown. This arrangement of the valves is exceedingly simple and allows very ready access to all three valves, as shown at fig. 50.

Ignition is accomplished by the low-tension electrical method. The magneto and tripping gear is clearly seen at figs. 48 and 49, and a section of the plug is shown as part of fig. 47. Fig. 51 shows a general view of the low-tension electric ignition gear, which shall be more fully described later, but here it may be stated that the magneto spindle directly operates the mechanical 'make and break' without intervening links, and the connection is so made that the sparking-plug can be withdrawn without disturbing any of the operative parts. The magneto lever is withdrawn and tripped by means of a tripping piece carried on the end of a straight rod which is reciprocated and rocked by means of the eccentric shown. This rod is carried in a swivelling sleeve which can be raised or lowered by means of a small lever, and this enables the ignition to be retarded or accelerated while the engine is running. Governing is accomplished by the method of 'hit or miss.' There are three main cams, inlet, gas, and exhaust, and an auxiliary or half compression cam for starting. The crank pin, the cylinder, piston, and piston pin are lubricated by a two-throw pump actuated by a lever resting on an eccentric cam on the side shaft. The oil container carrying the pump is seen at 52, fig. 48. Glass sight feeds, 61, are

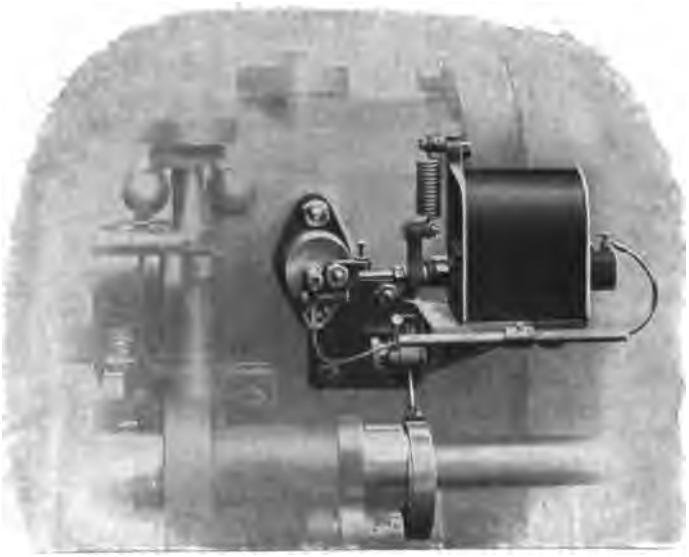
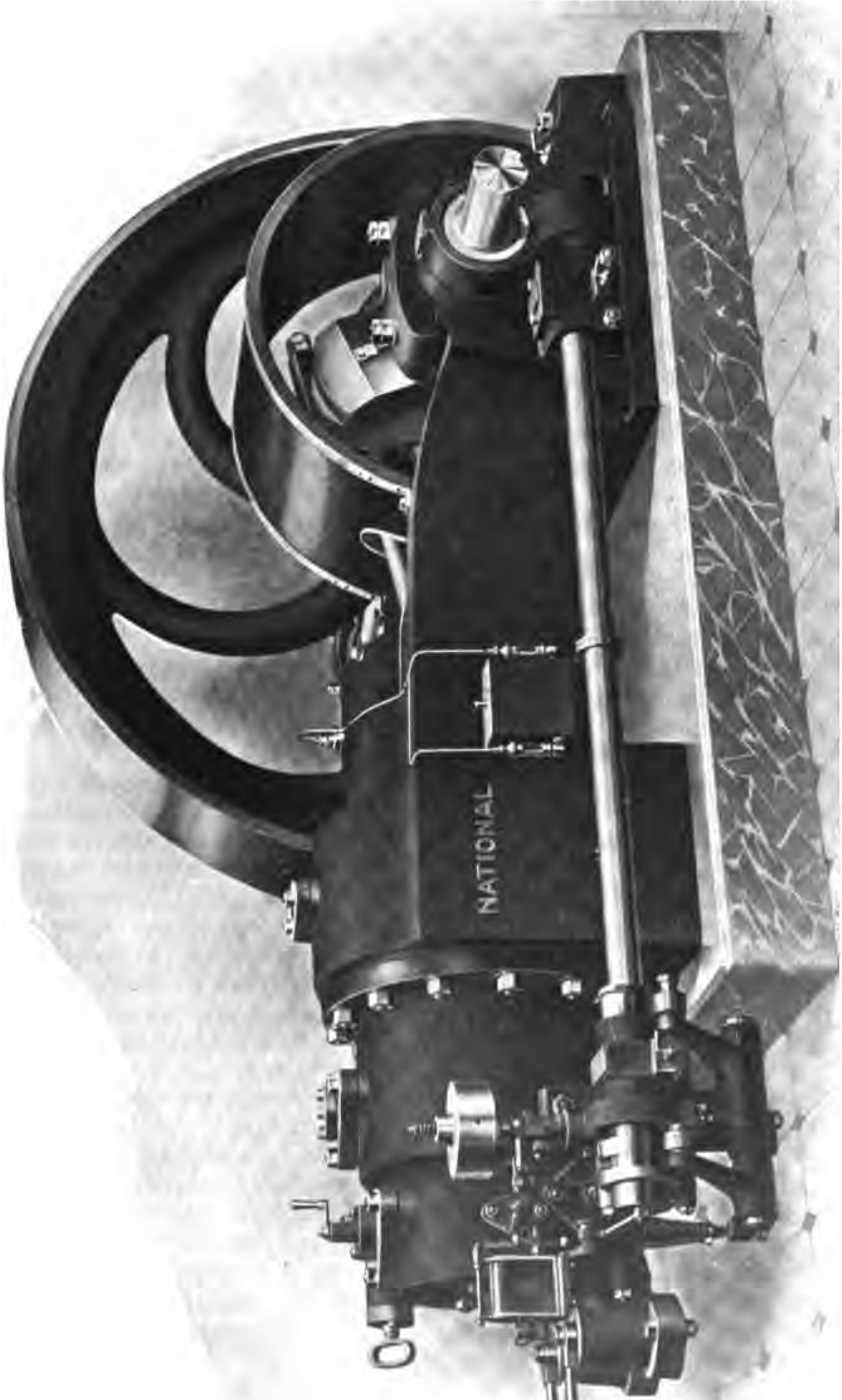


FIG. 51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fitted so as to enable the oil feed to be regulated. The oil passes to the liner by a pipe, and it lubricates cylinder and piston ; a portion, however, passes through the piston by a short tube and so reaches the piston pin 6. The oil from the other pipe passes into an open ring carried on the side of the crank, and reaches the crank pin through suitable pipe and bored passages by centrifugal action. This piston is cast with a circular ring, from which pass ribs to the piston walls. The exhaust, inlet, and gas valve levers are clearly seen in the illustration.

To start the engine the crank is set well off the centre on the working stroke, the starter valve is opened by the handle 55, the exhaust lever roller is set on the half compression cam, and a mixture of coal gas and air is pumped into the cylinder ; the electric ignition gear, which has been retarded, is then tripped by hand, and a compression explosion is obtained. The engine then draws in its charge of gas and air and ignites it at half compression until the speed rises. The starting valve is closed before the first explosion, and at speed the exhaust roller is moved into place for full compression. When no coal gas is available it is better to start with petrol vapour ; for this purpose a small cup is attached to the gas pump.

Fig. 52 shows a general view of the standard single cylinder horizontal engine of 16 ins. cylinder diameter and 22 ins. stroke.

Fig. 53 is a vertical longitudinal section of the engine, and fig. 54 is an external longitudinal elevation.

This engine differs but little from the smaller engine which has just been described. The air and gas valve casing is carefully water jacketed, and it and the combustion chamber casing are so constructed that on removing the former the combustion chamber water jacket is fully exposed so as to make it accessible for cleaning. As dimensions increase, so does the importance of the water jacket become greater ; in larger engines, therefore, it is necessary to clear out deposits likely to prevent the free flow of heat from the combustion chamber walls to the water. This arrangement requires a double joint, as seen at fig. 53 ; the inner joint is made with asbestos screwed up hard in the usual manner, and the outer joint is made with rubber so as to yield. Little pressure is required to hold the water joint, but great pressure is necessary to resist the explosion.

The spindle guides of all the main valves, gas inlet, and exhaust are separate from the casting, as clearly shown in the case of the exhaust valve spindle at fig. 53, to enable these guides to be removed after some years of wear. In the case of the exhaust valve it also permits a construction allowing of free and close access of water to the heated surface. The exhaust valve seating is also so placed as to permit of watering right up to the seat ; compare the section at figs. 47 and 53.

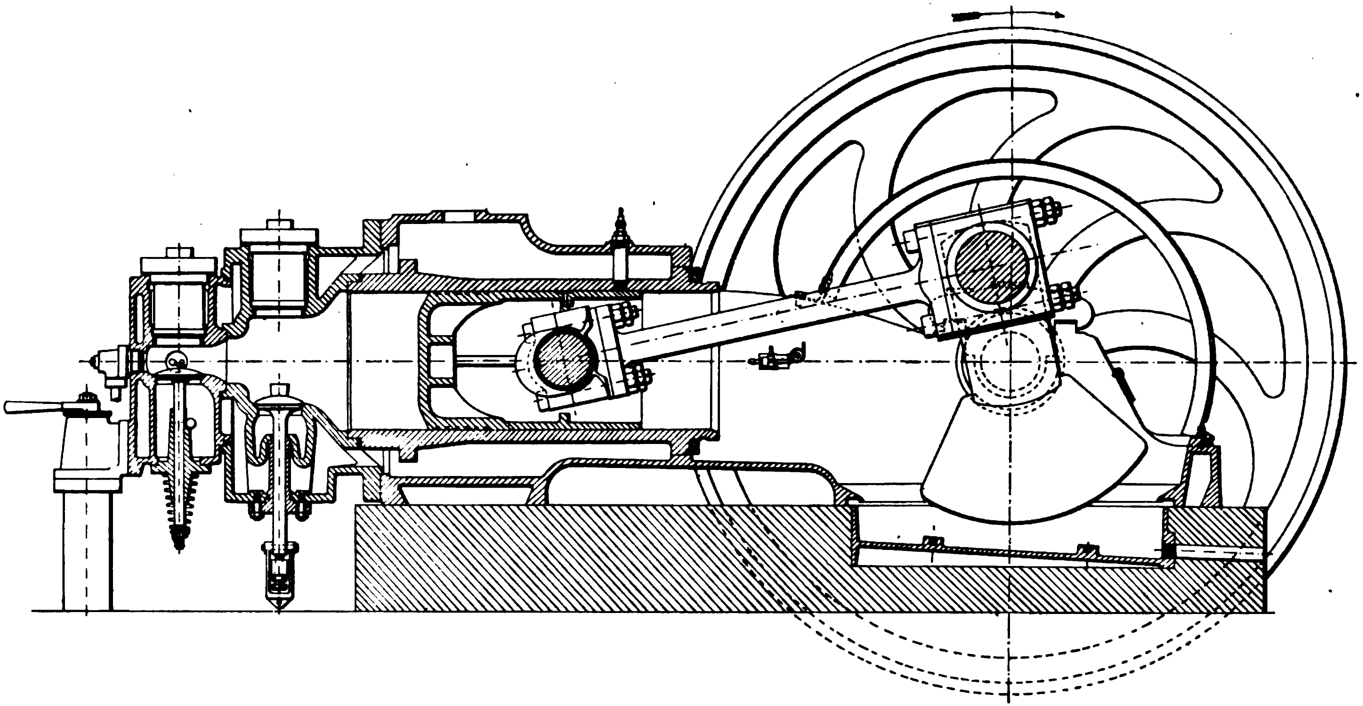


FIG. 53

The construction of the front expansion joint for the liner is also different, and the oil trough is cast separate from the bed, as shown. This lightens the large girder bed casting and yet provides for collecting the oil in such manner as to avoid saturating the foundation with oil.

The position of the magneto has been altered so as to allow operation of the tripping rod by a small crank pin in the end of the side shaft instead of an eccentric. Ring lubricators are also applied to the side shaft bearings as well as to the main bearings. Governing is effected by a variable admission gear, to be described later with reference to the larger engine. The gear varies the power of each impulse down to about quarter load, after which it cuts out ignitions.

Fig. 55 shows the general view of the standard single cylinder horizontal engine of 22 ins. cylinder diameter and 34 ins. stroke, and fig. 56 is a vertical longitudinal section.

The engine differs in two important particulars from the two smaller engines. Not only is the exhaust valve spindle guide separate from the combustion chamber casting, but the exhaust passage is carried in a separate water jacketed casting. The construction of this separate part is clearly seen at fig. 56. Here also there is an inner joint of metal to metal and an outer water joint. This arrangement simplifies and lightens the combustion chamber casting, a very important matter when castings become large, and complication tends to develop serious casting stresses.

The piston, too, is of a different construction, as is clearly seen in the section ; the bottom of the piston carries a ring which runs into the main piston casting beyond the rings ; apertures in the inner casting allow air to circulate. This construction offers two conducting paths from the hot centre of the piston bottom, and so keeps down the temperature below the point of pre-ignition.

In these larger engines two magnetos are fitted which operate separate ignition plugs ; fig. 57 shows three views of the arrangement adopted. The side shaft carries at its end a crank-pin, A, which operates a tripping rod, B ; this rod carries at its end a tripping piece, D, which engages with a hardened steel plate on the end of the magneto trip lever. O is the magneto ; P is the magneto spindle ; J the magneto trip lever keyed on to the spindle. This lever is three-armed ; the upper arm is connected to the spring H, which serves to return the spindle rapidly to its initial position after the trip mechanism lets go ; the second arm is J, and the third arm Q carries a striking pin, R, which operates the plug lever to secure a rapid break within the cylinder. K is the igniting plug, which contains a fixed contact pin, M, insulated by two porcelain plugs, and a movable spindle, N, carrying a contact making and breaking lever within the combustion port and an operating lever, S, outside ; this lever S is held back by the spring T.

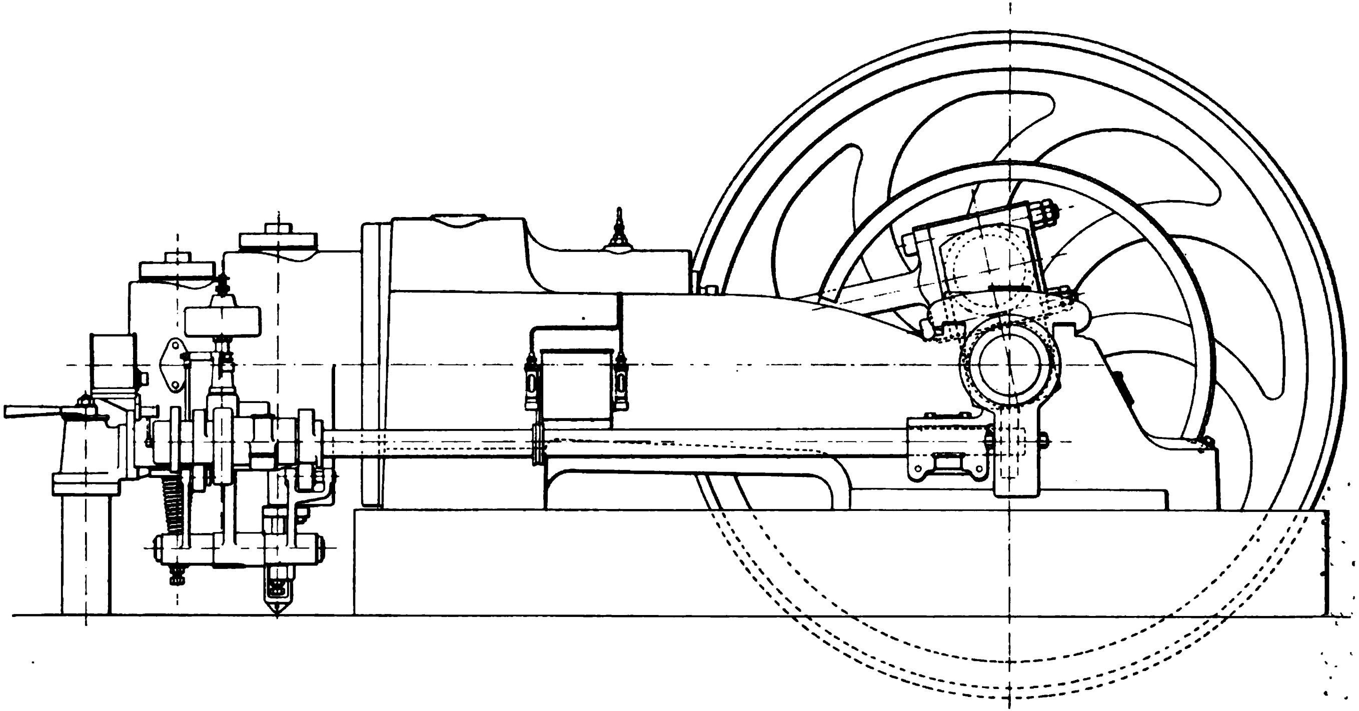


FIG. 54

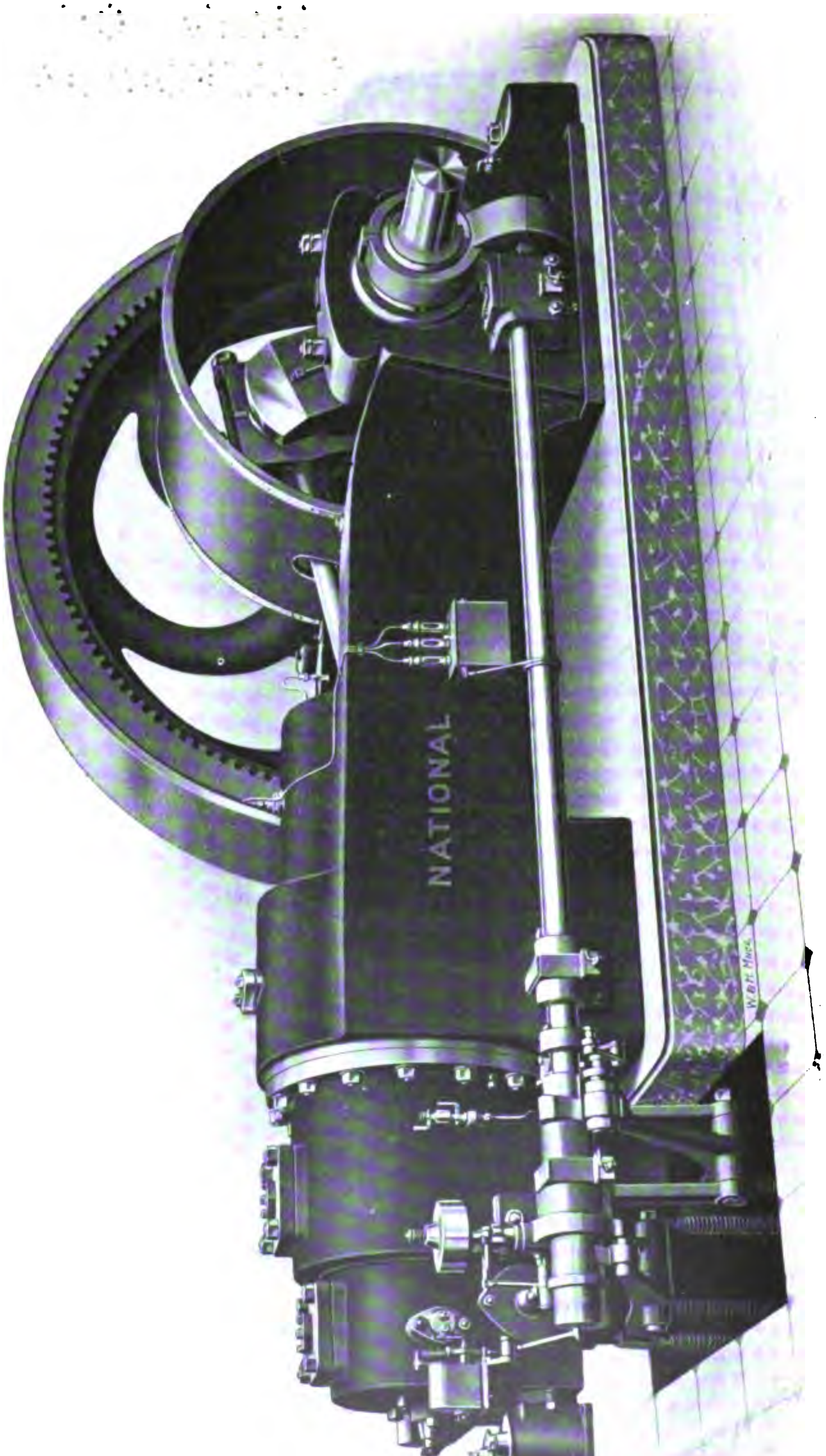


FIG. 55

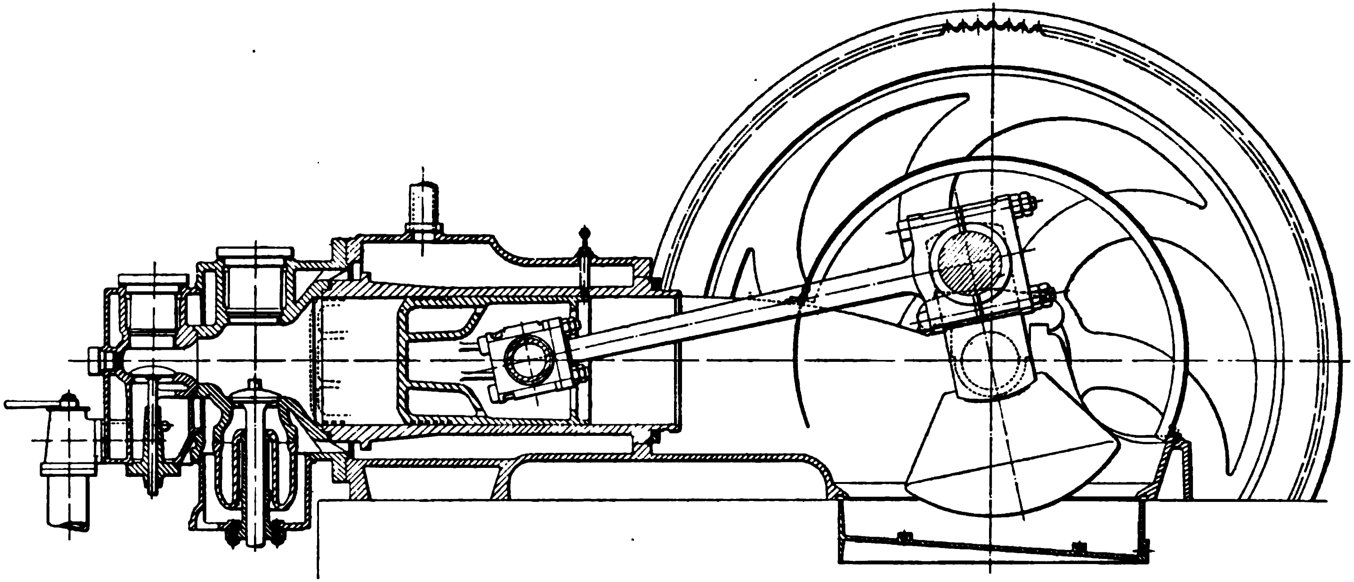


FIG. 56

The tripping rod B is carried in a sleeve, C, pivoted on the eccentric F, which can be moved by the timing lever U, and locked in any position

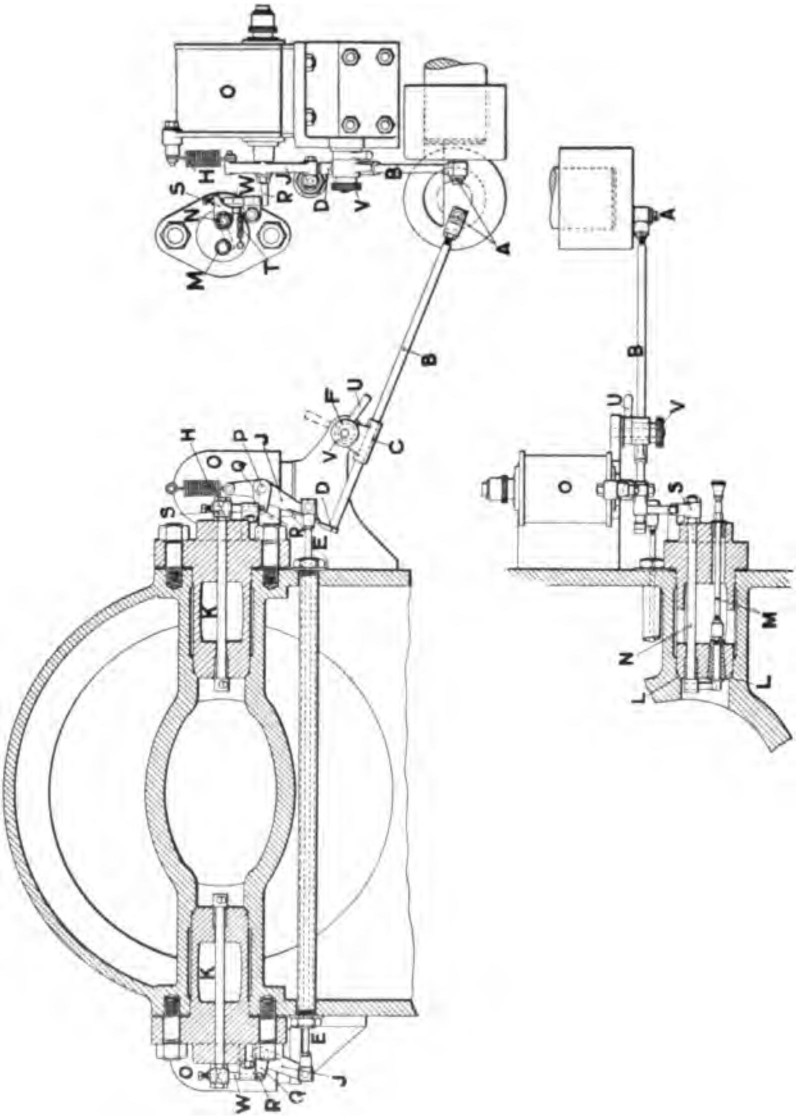


Fig. 57

by the milled head screw V. When the timing lever U is set in the position shown in full lines, the trip piece disengages the magneto lever soon and the ignition is early; when set in the dotted line position, it disengages late and gives late ignition. When the magneto lever is liberated by the trip piece, the spring H rapidly restores the shield or armature to its original position, and in doing so generates a low-tension current which passes through the closed contacts within the cylinder. Towards the end of the movement the striking pin R rapidly lifts the light short rod W, which lifts the contact operating lever S, and so rapidly breaks contact within the combustion port, and causes the spark which fires the charge.

An exactly similar magneto with a separate ignition plug is applied at the opposite side of the combustion port, and this second magneto system is operated from the magneto trip lever of the first by the connecting link E E, which passes within a tube carried through the water jacket; the link E E connects to the trip levers of the second magneto. In a modified arrangement the position of the magneto is altered so that the link E E passes outside the air and gas valve casing instead of through its water jacket. Such an arrangement is seen at fig. 55.

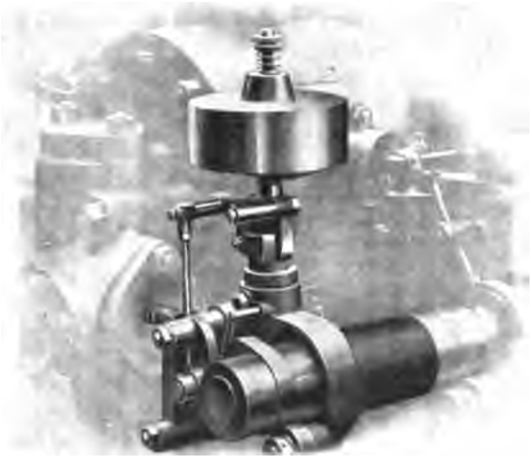


Fig. 58

The double ignition with separate plugs and magnetos greatly increases the reliability of the engine.

The variable admission governing gear depends partly on reducing the total weight of the charge and partly on opening the gas later and later with reference to the charge inlet valve and the air valve.

Fig. 58 shows a general view of the gear, and fig. 59 shows a transverse vertical section through the air and gas valve casing and an elevation of the governor and cut off operating gear. Two positions of the valves are shown; in the upper drawing the total lift of the air and gas valve spindle is ⅛ in., while in the lower figure the lift is ¾ in.; in the upper position the engine is almost without load, and in the lower position it is under full load.

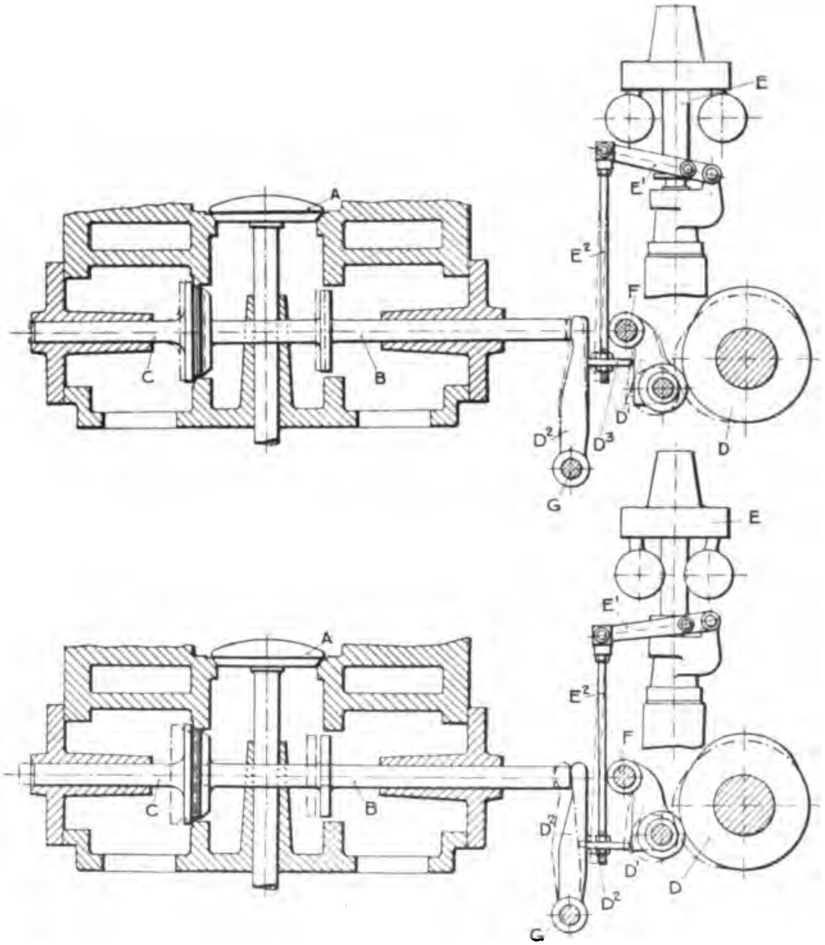


FIG. 59

The charge inlet valve A is opened and closed by its cam and lever at constant position at all loads, that is, it opens about 15° before the crank reaches its centre on the exhausting stroke, and it closes about 55° after the out centre on the compression stroke. The air valve B and the gas valve C are carried on the same spindle B C, but the air valve never closes on a seat; it is made of a smaller diameter than the air passage, so that even when the gas valve C seats entirely and completely prevents the admission of gas, the air valve or disc still permits air to pass to the main or charge inlet valve. The cam D on the side shaft operates the valve spindle B C by the agency of two levers, D^1 and D^2, and a contact or pusher plate, D^3. This plate D^3 is controlled as to its vertical position by the governor E and its lever and link E^1 and E^2. The lever D^1 is pivoted at F, and its roller is always held up against the surface of the cam D; it swings to and fro, therefore, from an upper fulcrum. The lever D^2 on the contrary is pivoted at G from below, and it is always held in contact with the valve spindle by a spring. When the cam D is in its out position the levers D^1 D^2 separate sufficiently to entirely liberate the pusher plate D^3, and at this period the governor is perfectly free to set the plate higher or lower according to its speed of rotation. In the high position shown in the upper drawing the plate finds itself near the fulcrum F, and so it can only receive a short stroke; it is also further away from the fulcrum G, and this also tends to diminish the stroke given to the spindle B C. In the lower position (lower drawing) the plate is farther from F and takes the full stroke (or a little more) of the cam, and the plate is also nearer the fulcrum G, and so the stroke given to the centre of the lever D^2 is nearly doubled at its extremity, where motion is given to the spindle B C. The lift is thus varied from ⅛ in. at light loads to ¾ in. at full loads. If all load be entirely thrown off the engine, then the governor lifts the pusher plate high enough to clear the acting surface of the lever D^1, as shown by the recessed part nearer to the pivot F. The cam D is so shaped that the cut-off point of the gas is nearly constant and early enough to allow the combustible mixture between the charge inlet valve A and the gas and air valves B C to be swept into the cylinder before A closes. This is necessary so as to fill the space with air, which then enters the cylinder on the charging stroke and cools down the exhaust gases before the admission of inflammable mixture. When the pusher plate is in its lowest position the full charge of gas and air is admitted throughout the whole suction stroke, and so the maximum diagram is obtained showing, with producer gas from anthracite, an effective pressure usually of about 80 lbs. per sq. in.; when the speed rises the gas supply is reduced, partly by later admission, partly by greater throttle. At first the compression is little reduced by the governing action, but the charge inlet port and the combustion chamber are filled with inflammable mixture, while the cylinder itself receives less or may even contain little but air. The ignition then produces a reduced diagram, although the charge weight drawn in is but little affected. At the lighter loads the air inlet valve or disc B does not move enough to open the air passage in which it works; that is, the air passage consists only of the annulus formed by the disc in the bored out port, and the weight of the charge is thus reduced without the proportion of gas being greatly altered.

Reproductions of indicator diagrams taken by the author at Ashton-under-Lyne from engine No. 22597, using Mond gas, on July 12, 1912, are given in figs. 60, A, B, C, D. This engine had a cylinder of 20 ins. bore, and a stroke of 31 ins., and ran throughout at 170 r.p.m. The full load diagram, fig. 60 A, shows a mean effective pressure of 90·1 lbs. per sq. in., and the lower load cards B, C, and D show respectively M.E.P.s of 77·4, 67·3, and 55·6 lbs. per sq. in. Missed working strokes are indicated at M M on figs. 60 B and D.

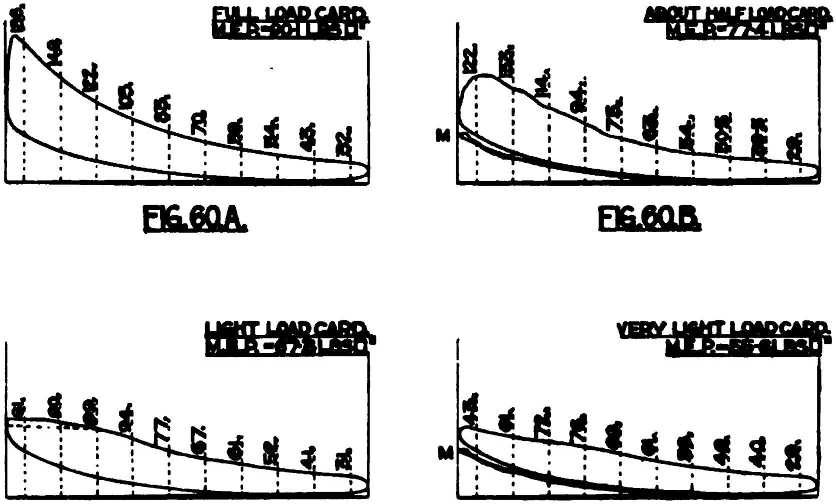


FIG.60A. FIG.60B.

FIG.60C. FIG.60D.

FIG. 60

But little change in compression pressure occurs in the series of diagrams here shown; the full load card shows a compression pressure of 130 lbs. per sq. in., the other three showing each about 122 lbs. per sq. in.

The maximum explosion pressure in the full load card is 302 lbs. per sq. in. above atmosphere, while in fig. 60 C it can hardly be said that an explosion occurs, the maximum pressure indicated being only

160 lbs. per sq. in. above atmosphere, and this remains practically constant during about one-fifth of the working stroke.

These diagrams show that the National Company's variable impulse governor acts well, while the illustrations given show that these favourable working results are attained by very simple means.

The larger engines are started by compressed air.

Tests of Power and Fuel Consumption

The tests of the National engines by the Institution of Civil Engineers Committee on the Standards of Efficiency in Internal Combustion Engines are the most authoritative of the numerous tests which have been made with the National Company's gas engines. Many matters were tested beyond those required for commercial trials, and they have been discussed in the first volume of this book. The leading particulars and results are given in the following table:

Tests of Three National Gas Engines having the same Compression Ratio

(*Committee, Institution of Civil Engineers*). 1905

	Dia. Stroke	Dia. Stroke	Dia. Stroke
Dimensions	5·5 ins. × 10 ins.	9 ins. × 17 ins.	14 ins. × 22 ins.
Mean revolutions per minute during test	258·9	203·6	165·8
Brake horse-power	5·2	20·9	52·7
Brake thermal efficiency	26·7%	28·3%	29·8%
Coal gas consumption per BHP hour at working temperature and pressure	16·87 cub. ft.	15·84 cub. ft.	14·9 cub. ft.
Lower calorific value of gas at working temp. and pressure	566 B.Th.U.	567 B.Th.U.	574 B.Th.U.
Ratio by volume air to gas	$\frac{\text{air}}{\text{gas}} = \frac{9·15}{1}$	$\frac{\text{air}}{\text{gas}} = \frac{9·17}{1}$	$\frac{\text{air}}{\text{gas}} = \frac{8·21}{1}$
Ratio by volume air + exhaust to gas	$\frac{\text{air \& exhaust}}{\text{gas}} = \frac{10·1}{1}$	$\frac{9·75}{1}$	$\frac{9·3}{1}$
Volume of combustion space	52 cub. ins.	239 cub. ins.	774 cub. ins.
Volume swept by piston	237·8 cub. ins.	1083 cub. ins.	3390 cub. ins.
Total volume of cylinder	289·8 cub. ins.	1322 cub. ins.	4164 cub. ins.
Compression ratio	5·55	5·52	5·38
Air standard efficiency	49·6%	49·6%	49%

The Committee took indicator diagrams, but they considered the brake results to be the more accurate and so have given brake thermal efficiency and not indicated. The author, however, has determined the mechanical efficiency by two methods which give as mean results from small to large engine 84 per cent., 85 per cent., and 86 per cent., and these figures give corresponding indicated thermal efficiency as 31·8 per cent., 33·3 per cent., and 34·7 per cent.

These tests prove conclusively that the increase of dimensions has but a small effect in improving economy, and that further, with a cylinder of only 14 ins. diameter, having the moderate compression ratio of 5·38 or about 125 lbs. per sq. in. above atmosphere, a brake thermal efficiency of practically 30 per cent. may be obtained, and an indicated efficiency of nearly 35 per cent. The highest efficiency values, however, are obtained when the mixture is somewhat weaker than that required to produce maximum power. In ordinary work maximum power is usually desired, and this is obtained by using rather richer mixtures which cause a small fall in thermal efficiency.

The results obtained in ordinary practical work are well shown in tests made by Messrs. Hall, Williams, and Brydges in 1909 at the works of Messrs. Rowan & Co., engineers, of Glasgow. These works are driven by four National suction gas engines of the single cylinder horizontal type, each cylinder being 22½ ins. diameter by 30 ins. stroke, and the revolutions 170 per minute. Governing was by the hit or miss method.

The average results were as follows :

	Feb. 25, 1909.	Feb. 26, 1909.
Average BHP developed by each engine . . .	117·6	136
Average fuel consumption per BHP hour (Scotch anthracite), lbs.	0·854	0·845
Cooling water consumption, gallons per BHP hour .	4·2	3·64

Each engine was rated at 150 BHP, and each was capable of carrying a heavy overload for long periods. Scotch anthracite was used in four suction producers, and its cost at the date of the trials was 18*s*. per ton, delivered at the works.

The lower thermal value of Scotch anthracite is about 13,500 B.Th.U. per lb. Taking 0·845 lb. per BHP as the best result, and assuming 80 per cent. as the efficiency of the producer, gives the brake thermal efficiency of the engines as 28 per cent.—a very satisfactory value.

Many other tests show that the smaller National engines, say of about 30 BHP, with suction producers use 1 lb. of Welsh anthracite per BHP hour at full load, while larger engines of, say, 150 BHP use 0·75 lb. of Welsh anthracite per BHP, also at full load.

Of these, three examples will be given illustrating the results of running in ordinary commercial work with coal gas and anthracite producer gas.

The following test was made with a twin cylinder engine working on coal gas at Darlington with the load about three-quarters, the indicated HP 334·5 and the BHP 276:

TEST OF A NATIONAL GAS ENGINE, TWIN CYLINDER, SINGLE-ACTING HORIZONTAL TYPE, WORKING ON COAL GAS AT THE CLEVELAND BRIDGE AND ENGINEERING CO., LTD. *Darlington. August* 16, 1910

Two cylinders, each 22 *ins. diameter and* 30 *ins. stroke. Duration of test* 1 *hour.*

	Cylinder No. 13267.		Cylinder No. 13268.
Total revolutions in 1 hour	9491		9491
Average revolutions per minute	158·18		158·18
Total explosions in 1 hour	3555	and in 45 min.	3006
Average explosions per minute	59·25		66·8
Average mean pressure, lbs. per sq. in.	92		92·65
Indicated HP	157		177·5
Total indicated HP			334·5
„ brake HP			276·0
Mechanical efficiency			82·5 per cent.
Total coal gas used for 1 hour at 71·5° F. and 29·6 ins. bar.			4832 cub. ft.
„ „ „ „ „ „ corrected to 32° F. and 30 ins. bar.			4420 cub. ft.
Coal gas consumed per BHP hour			16 cub. ft.
„ „ „ „ kilowatt hour			23·2 cub. ft.
Calorific value (lower) at 32° F and 30 ins. bar.			540 B.Th.U. per cub. ft.
Brake thermal efficiency			29·7 per cent.
Indicated thermal efficiency			36·0 per cent.

The brake thermal efficiency is practically the same as that obtained for the 14 ins. engine by the Institution of Civil Engineers Committee, and it proves that 30 per cent. is by no means unusual at full load.

In the following test the engine is operated by suction producer gas from anthracite:

TEST OF A NATIONAL GAS ENGINE OF SINGLE CYLINDER, SINGLE-ACTING HORIZONTAL TYPE, WORKING WITH SUCTION PRODUCER AND ANTHRACITE AT MESSRS. STUBBS & CO. *Manchester. October* 23, 1900

Cylinder 18½ *ins. diameter and* 24 *ins. stroke. Duration of test* 8 *hours.*

Total revolutions in 8 hours	95273
Average revolutions per minute	198·49
Average mean pressure	77 lbs. per sq. in.
Explosions per minute	91·03
Indicated HP	114
Brake HP	91
Kilowatts	60
Total anthracite consumed in 8 hours	598 lbs.
„ „ „ per hour	74·7 lbs.
„ „ „ per IHP hour	0·655 lbs.
„ „ „ per BHP hour	0·822 lbs.
„ „ „ per kilowatt hour	1·246 lbs.

The brake thermal efficiency of this engine at ¾ load is about 27 per cent.

The following test was made with producer gas during two working weeks at the ordinary work of the factory, and extended to 118·66 hours or 59·33 hours per week. The engine was capable of a maximum brake power of 130 horse, and the average load during the time run was 62·9 BHP, so that only half load was required. The fuel used was anthracite, and the total includes standby losses during the night and fuel used for lighting up the producer at the beginning of each week:

TWELVE DAYS' TEST OF A NATIONAL GAS ENGINE OF SINGLE CYLINDER, SINGLE ACTING HORIZONTAL TYPE WHILE AT WORK AT THE GREENBANK IRONWORKS, BLACKBURN. *January* 13 *to* 19 *inclusive and January* 21 *to* 26, 1907, *inclusive*

Cylinder 19 *ins. diameter and* 28 *ins. stroke. Max. BHP,* 130.

Engine No. 13140—Worked by Suction producer with Anthracite.

Duration of test 12 days	118·66 hrs.
Average revolutions per minute	164–166
Average explosions per minute	42·2–48·1
Average BHP 1st week, 58·83 hrs.	62·4
„ „ 2nd week, 59·83 hrs.	62·8
Total BHP hours	7435
Total anthracite consumed in 118·66 hrs.	8396 lbs.
Anthracite consumed per hour	70·75 lbs.
Anthracite consumed per BHP hour (including standby and lighting up losses for two weeks)	1·12 lbs.
Lubricating oil used in engine during 59·83 hrs. :	
On piston 6·5 pints On crankshaft 19·5 pints	26 pints total
Lubricating oil used on piston per BHP hour	0·0017 pints
„ „ „ „ crankshaft „ „	0·005 pints
Coal gas used for starting engine in 12 days	440 cub. ft.
Water used for gas plant only during 12 days' test of 118·66 hrs. per BHP hour	1·54 gallons

In this producer the standby losses during the night and the meal hours amount to about 4 lbs. of anthracite per hour. Allowing for this and for fuel required for lighting up at the beginning of each week, the running consumption for about half load is just under 1 lb. per BHP hour.

From this test the following costs for fuel, oil, water, and coal gas have been taken for one week's run:

COST OF FUEL, LUBRICATING OIL, WATER, AND COAL GAS FOR ONE WEEK'S RUN AS ABOVE

	£	*s.*	*d.*
Water used in one week	0	6	0
Cylinder oil	0	1	1
Bearing oil	0	4	5½
Anthracite used, at 28*s.* 6*d.* per ton	2	12	8
Coal gas, at 3*s.* per 1000 cub. ft.	0	0	8
Total cost fuel, oil, water, and coal gas for one weeks' run	3	4	10½

Total hours run per week	59·33
BHP	62·9
Total BHP hours	3725
Cost of fuel, oil, water, and coal gas per BHP hour	0·209*d.*

Cylinder oil cost	1*s.* 4*d.* per gallon
Bearing oil „	1*s.* 10*d.* „
Anthracite „	28*s.* 6*d.* per ton
Coal gas „	3*s.* 0*d.* per 1000 cub. ft.

These tests give a sufficient idea of the results to be expected from the National Co.'s horizontal engines under both scientific and workshop conditions.

OIL CONSUMPTION

The lubricating oil consumed in these engines is approximately as follows:

Cylinder diameter.	Lubrication oil consumption.
4 ins., 4·5 ins., and 5 ins.	1 pint for 10 hours' run
5½ ins., 6·5 ins., and 7 ins.	1·75 pints „ „ „
7 ins., 8 ins., and 9 ins.	2·5 „ „ „ „
10 ins., 11 ins., and 12 ins.	3·25 „ „ „ „
13 ins., 14 ins., and 16 ins.	4 „ „ „ „
17 ins., 18·5 ins., 20 ins., and 22 ins.	5 „ „ „ „

WATER CONSUMPTION

If the cooling water be run to waste, 3·5 gals. are required per BHP hour with water leaving the engines at about 140° F. Where tanks or coolers are used, however, the evaporation only requires to be made up, and this amounts to about 0·2 gallons per BHP hour. An engine of 20 ins. cylinder × 31 ins. stroke requires a tank capacity for storing and cooling of 5000 gallons.

The suction producer gas plant requires for cooling and scrubbing purposes 1 gallon of water per BHP hour, and this water in most cases should be run to waste.

CONSUMPTION OF ANTHRACITE, COKE, AND CHARCOAL IN SUCTION PRODUCERS

Under ordinary workshop conditions the following figures give roughly the consumption per BHP hour of Welsh anthracite, Scotch anthracite, coke, and charcoal which can be readily obtained in good commercial suction gas producers working with an engine of about 150 max. BHP at various loads.

Fuel	Full load	Three-quarter	Half	Quarter load
Welsh anthracite . .	⅞ lb.	0·95 lb.	1⅛ lbs.	1⅜ lbs.
Scotch anthracite . .	1 lb.	1·1 lbs.	1¼ lbs.	—
Coke	1¼ lbs.	1⅜ lbs.	1⅝ lbs.	2¼ lbs.
Charcoal	1⅛ lbs.	—	1⅜ lbs.	1¾ lbs.

The lower thermal values of these fuels are approximately :

Welsh anthracite	14,000 B.Th.U. per lb.
Scotch anthracite	13,500 „ „
Coke	10,000 „ „
Charcoal	12,700 „ „

OTHER BRITISH GAS ENGINES

Many British engineers build excellent four-cycle gas engines. Among those who are longest established may be mentioned the Stockport Engine Co., the Premier Gas Engine Co., Tangye, Ltd., Fielding & Platt, and the Campbell Gas Engine Co., Ltd. Of these the Premier Co. have long adhered to the positive air scavenging type, and have paid special attention to large power engines.

An early scavenging engine is described in the 1896 edition of this work as follows :

'Engines of 20 nominal HP and above are made with a scavenging arrangement to displace the exhaust products. The front end of the piston is enlarged and forms an annular cylinder which serves as an air pump. On the return stroke the air is discharged from the annulus, and passed through the combustion space of the cylinder so as to displace the burnt gases by pure air. For ordinary work the engine is made with a single cylinder, but for electric lighting two cylinders are used arranged in tandem. Fig. 61 shows in elevation an engine of the tandem type capable of indicating 120 HP with Dowson gas. The engine is constructed and operates as follows. The front motor piston has a large end which works in the bored bed plate ; to this end the connecting-rod is attached, so that it acts as a guide block.

Two side rods are secured to the end and passed backwards alongside the cylinder liner through a passage cast in the water jacket: thence they pass through bushes having light spring rings and secured at

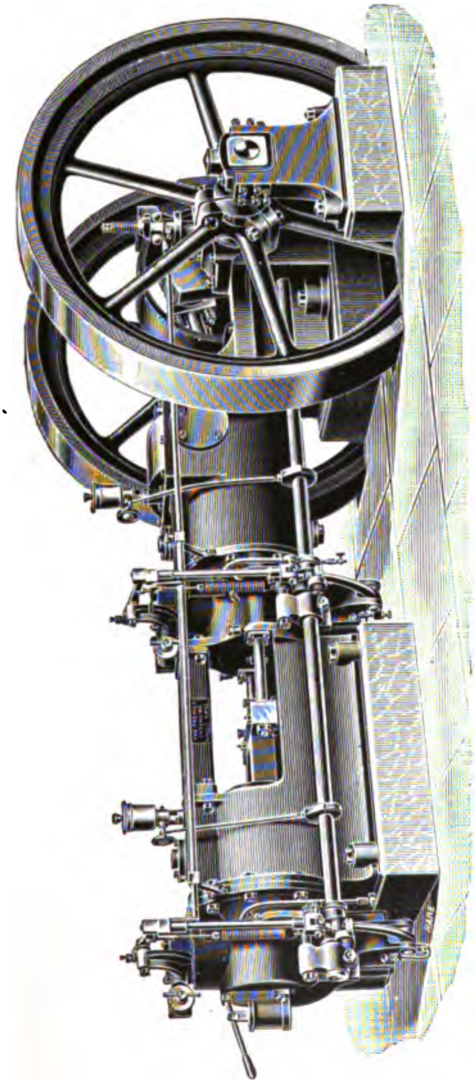


FIG. 61.—Wells Otto Engine Junction Cylinders, 60 BHP

their rear ends to the crosshead of the back piston. The large piston acts as an air-pump, but a free passage to the atmosphere is provided during the first part of the back stroke, so that the air intended for

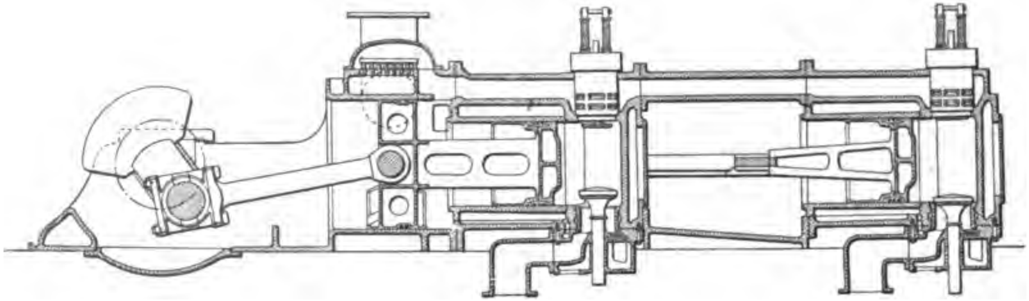
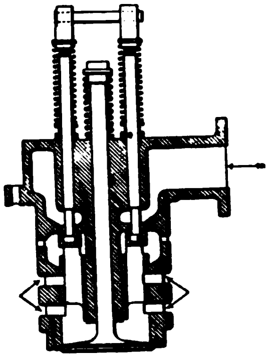


Fig. 62



scavenging is only compressed and passes through the combustion chamber towards the end of the exhaust stroke. As the cylinders make exhaust strokes alternately, and the large piston forces air through the air passage leading to both cylinders at every back stroke, the air is discharged through whichever of the motor cylinders is on its exhaust stroke.

'The governor is of the high speed spring loaded centrifugal type, and is driven by a bevel wheel on the crankshaft; it controls upright hit and miss rods in such a manner that the gas is cut out from one cylinder before the other. The proportion of gas admitted is also varied between narrow limits by graduated notches, which determine the lift of the gas valves. The engine has two flywheels and outside adjustable bearings, positive ratchet feed lubricators for the cylinders, an oilbox on the splash guard, and sight drop feed supply to the main bearings and to the crank pin by a centrifugal oiler.'

Mr. Humphrey, in his paper[1] on power gas and large gas engines of 1901, describes a larger Premier engine of the tandem type, of which fig. 62 is a longitudinal section. It is thus described by Mr. Humphrey.

'The engine was made by the Premier Gas Engine Co., Sandiacre, near Nottingham, for Messrs. Brunner, Mond & Co. It is a two-cylinder engine of the positive scavenger type, arranged with the cylinders tandem fashion and having a third piston which serves the double purpose of pumping the scavenging air and acting as a guide or crosshead of large bearing surface. The motor pistons are of the Premier water-cooled type, and side rods are used to couple the back piston to the crosshead. Only one connecting-rod is required, and impulses are received from each motor cylinder alternately. The engine is direct connected by means of a "cheese" coupling to a Mather and Platt dynamo, having a normal output of 2250 amperes at 110 volts.'

Mr. Humphrey gives the following test of this engine:

RESULTS OF TESTS MADE WITH 500 HP GAS ENGINE (PREMIER) WORKING WITH MOND GAS. *Winnington Power House. December* 7, 1900

Trial with Engine giving Two-thirds of its Maximum Output.

Duration of test, 12.30 p.m. to 5.30 p.m. . . .	5 hours
Dimensions of engine:	
Two cylinders, arranged tandem, each .	28⅛ ins. diam.
Pump cylinder, for scavenging air . .	43½ ins. diam.
Length of stroke	30 ins.

The engine is direct coupled to a Mather and Platt dynamo.

Average revolutions per minute	128·05
Load on engine, as fraction of maximum output . . .	⅔
Number of explosions per minute, back cylinder . . .	64·02
" " " front "	31·75

---

[1] 'Power Gas and Large Gas Engines,' Inst. Mech. E., 1901.

Mean effective pressure, average, for back cylinder	109·1	lbs. per sq. in.
„ „ „ „ „ front „	107·4	„ „ „ „
„ „ „ „ „ pump	2·5	„ „ „ „
„ „ „ ‘ bottom loop ’ back	2·25	„ „ „ „
„ „ „ „ „ front	2·63	„ „ „ „
Indicated horse-power, back cylinder	328·72	
„ „ front „	160·49	
„ „ gross total	489·21	
„ „ pump	21·39	
„ „ ‘ bottom loop ’ back . .	6·77	
„ „ „ „ front . .	7·83	
„ „ total fluid losses	35·99	
Output of dynamo, average amperes	2320·0	
„ „ „ volts	110·1	
Electrical horse-power	342·4	
Kilowatts	255·43	

Efficiencies :

Electrical efficiency of dynamo	93	per cent.
Mechanical efficiency of engine, excluding fluid losses .	81·22	„ „
Brake HP at dynamo coupling	368·2	„ „
Total IHP lost, including all frictional, fluid, and pump losses	121·0	„ „
Combined efficiency, EHP/IHP, per cent. . . .	70·0	„ „
Average temperature of jacket water, back . . .	49° C.	
„ „ „ „ „ front . .	37° C.	
Gas used—total as measured by meter	136,100	cub. ft.

Temperature of gas, 14·6° C. Barometer, 29·992 ins.

Dry gas at 0° C. and 760 mm. used per hour . . .	25,482	cub. ft.

Analysis of Gas :

CO_2 . . .	16·0	vol. per cent.
CO . . .	12·2	„ „
H . . .	27·8	„ „
CH_4 . . .	2·2	„ „
N . . .	41·8	„ „

Total combustible gases, volume per cent.	42·2
‘ Higher ’ calorific value, including latent heat of steam :	
Kilo-calories per cubic metre, 0° C.	1432·7
British thermal units per cubic foot	160·9
‘ Lower ’ calorific value, excluding latent heat of steam :	
Kilo-calories per cubic metre, 0° C.	1,280·4
British thermal units per cubic foot	143·8

	Cub. ft.	Cub. m.
Mond gas at 0° C. per IHP hour	52·09	1·475
„ „ „ „ BHP hour . . .	69·20	1·959
„ „ „ „ EHP hour . . .	74·42	2·107
„ „ „ „ Board of Trade unit .	99·76	2·825

Thermal efficiencies	Calculated on ‘ higher ’ calorific value	Calculated on ‘ lower ’ calorific value
	Per cent.	Per cent.
Calculated on the IHP	30·38	34·00
„ „ BHP	22·87	25·59
„ „ EHP	21·27	23·80

The author inspected this engine in 1902, and found it running smoothly with a heavy load.

Fig. 63 is a diagram taken from the engine at the time of his test by Mr. Humphrey.

Many large engines have been constructed by the Premier Company with open trunk cylinders, but their later engines use a piston rod and stuffing-box to connect the tandem pistons instead of side rods and crosshead. The weight of the reciprocating parts is thus diminished and higher piston speeds are made possible. The type in fact resembles the Crossley engine of fig. 40, but provision is made for an additional cylinder for scavenging. Such an engine is shown in section at fig. 64, where the pistons are supplied with water from the swinging connection shown between the cylinders; pistons and piston rod are watered effectively. The inlet and exhaust valves are arranged horizontally and they open right into the combustion space without port or passage between valve and cylinder.

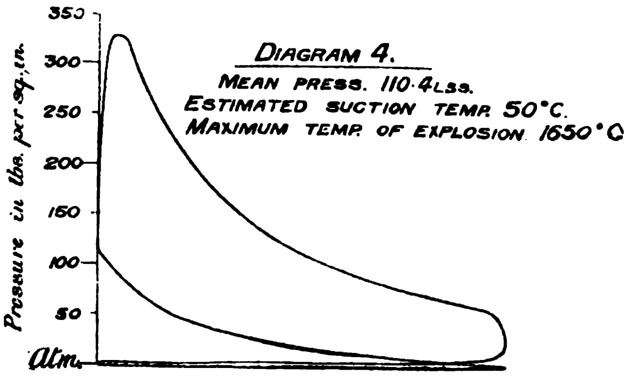


Fig. 63

The scavenging air is supplied to the cylinders by the pump mounted above whose piston is actuated from the main crank by means of a light connecting-rod working on an extension pin from the main connecting-rod-brasses. This arrangement has been extensively applied by the Premier Company. Altogether they have supplied nearly 57,000 horse-power with cylinder diameter up to 39 ins. and stroke 48 ins.

The design is good, although the tandem double-acting engines to be described later in the chapter may now be regarded as standard practice on the Continent, in America, and England.

With regard to this engine Mr. Allen states:

'Varied views are held as to the advantages to be derived from more or less completely scavenging out the products of combustion with cool air before the mixture is admitted. In the Premier engine illustrated in fig. 64 it will be seen that a separate air pump driven off the big end of the connecting-rod is used to supply the air to the scavenging charge, and Mr. Hamilton, the designer of these engines.

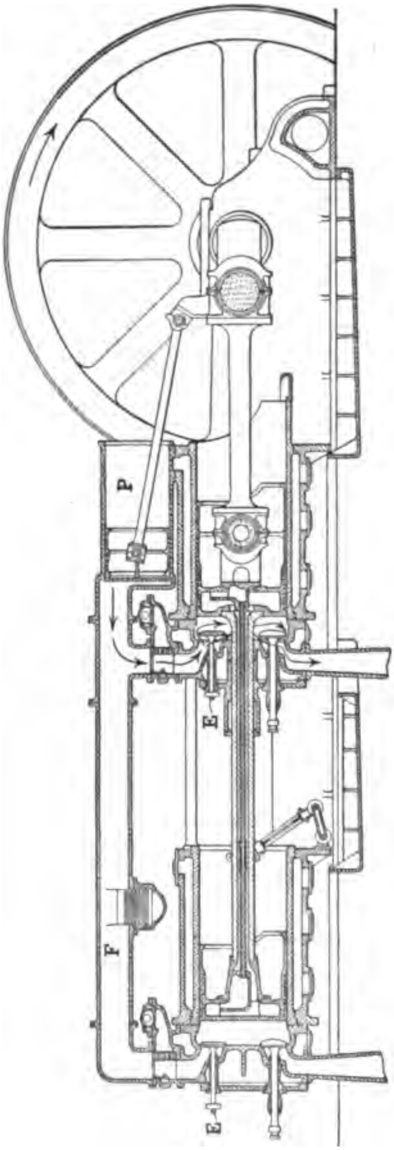


FIG. 64

has always consistently held that a positive scavenge of this kind is advantageous, the argument being that the interior of the combustion

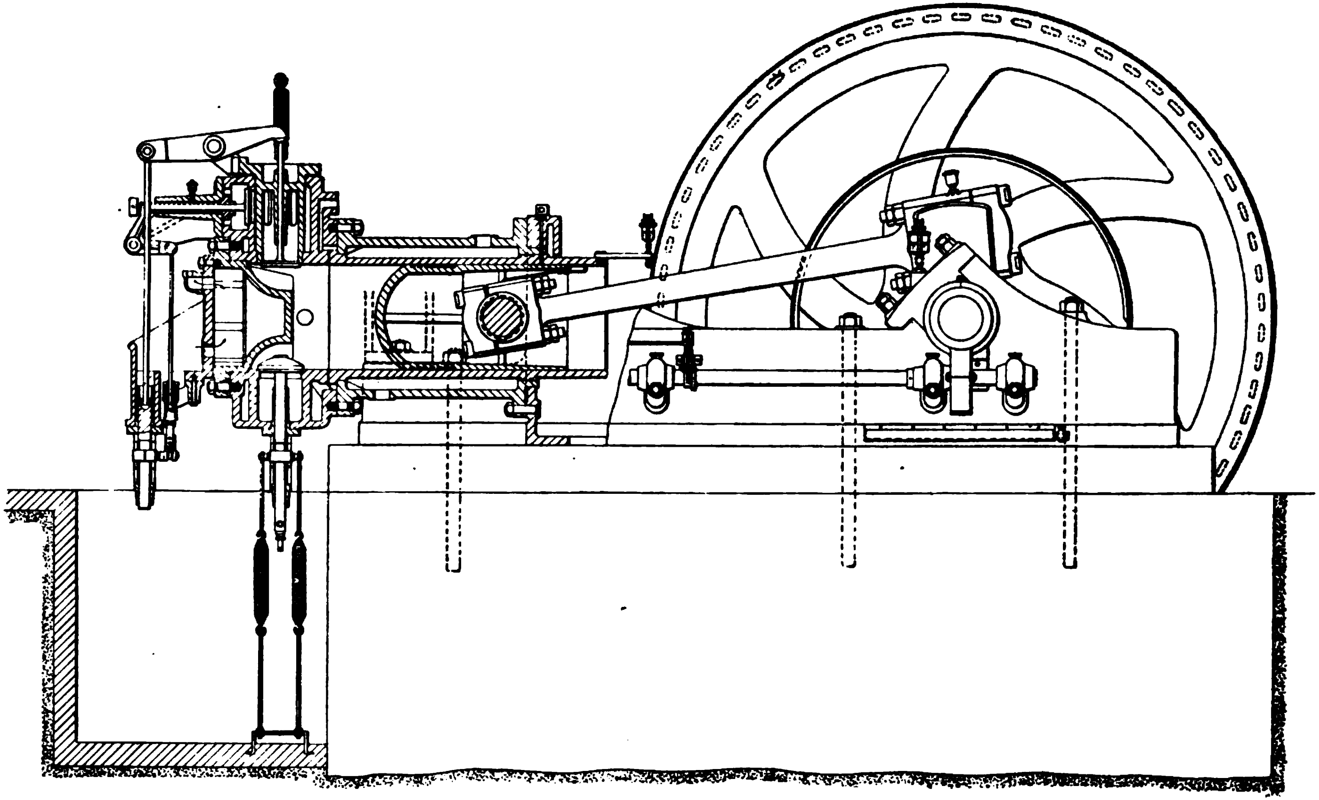


FIG. 65

chamber is effectively cooled and the lingering products of combustion likely to cause pre-ignition are effectively expelled, and that with this arrangement the engine will develop more power for a given amount of gas, even after taking into account the extra friction of the air pump, which is estimated at 2 to 3 per cent. of the total power developed, and there seems no doubt that a positive scavenge permits a higher degree of compression than when a portion of the products of combustion is left in.'

A recent Stockport engine is shown in longitudinal section at fig. 65. The cylinder is 20 ins. diameter and the stroke is 30 ins.; at 160 revolutions per minute this engine gives as a maximum 200 BHP when consuming coal gas. The design of the engine is simple and effective. The cylinder is free at the front end and the water joint is made by rubber rings.

The cylinder consists of a hard liner with a flange at its rear end, of which part fits into the bored out facing forming part of the water jacket casing and engine frame; the flanged part projects over the end facing, and it is held in position by the breech casting, which is bolted on the water casing flange by studs and nuts. This requires the making of a double joint, but the inner joint which takes the explosion pressure is hard, while the outer joint which requires to resist water pressure only is soft, so that there is no difficulty in keeping both joints pressure tight. The breech end is cast as a symmetrical ring hollowed for water and cored out for the inlet and exhaust valves, which are arranged top and bottom—inlet valve above and exhaust below. The breech is closed by a hollow cylinder cover, which projects into the cylinder past the centres of the opposed valves, where it is hollowed out to allow of lift. The valves thus open directly into the cylinder practically without ports, and high compressions are rendered possible. It is found that in engines of 20 ins. cylinder and above it is better to retain a considerable cooling surface; it is easy in large engines to reduce the proportion of cooling surface to total combustion space volume to such an extent as to produce frequent pre-ignition. To avoid pre-ignition it is necessary either to increase cooling surface or to scavenge out the cylinder with air or cool gases. In some cases the cooling is accomplished by injecting water spray into the charge. The exhaust valve port and seat are arranged in the casting, but the exhaust spindle sleeve or guide is made in a separate piece. It is somewhat peculiar, however, to find that the Stockport Company do not find it necessary to water this sleeve, as is done by most other first-class makers. They instead water the exhaust valve and its spindle, which is probably quite as effective.

The inlet valve is carried in a cage with its seat, and the cage seats against a ground joint in the usual manner. To get out the exhaust valve, it is necessary to remove the cylinder cover and lift out the inlet valve with its cage.

The gas supply valve as shown is arranged at right angles to the inlet valve, the gas mixing with the entering air by passing through slots in the cylindrical wall of the cage.

The arrangement for operating the gas valve from the inlet pusher is clearly indicated.

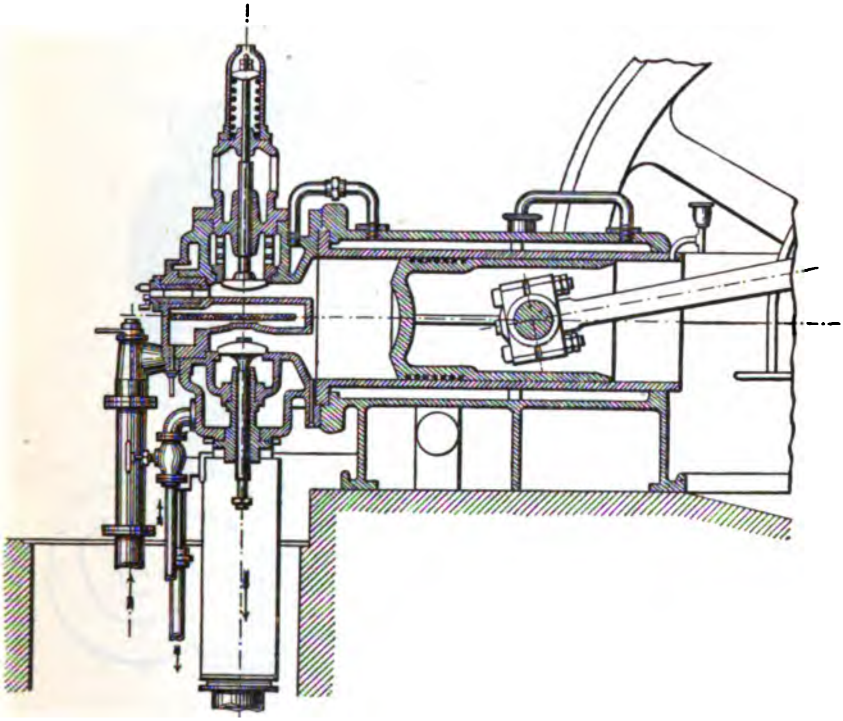


Fig. 66

The piston is of the air-cooled type, and it is heavily domed at its hot end to allow freedom for expansion at the centre without unduly stressing the periphery.

This engine, like all modern engines, undoubtedly gives high thermal efficiencies, depending on the compression ratio which it is thought desirable to adopt. Referring again to the ratio of cooling surface to cylinder volume, it is interesting to glance at fig. 66, which is a longitudinal section of a Koerting four-cycle engine of 250 HP. Here the difficulty of over-heated combustion chamber and too hot exhaust products is very frankly met. A projecting tube has been deliberately inserted into the combustion space and provided with water circulation to increase the rate of cooling of the exhaust products. Such a device is, of course, bad from the point of view of maximum heat economy, but it enables a relatively large cylinder engine to operate successfully without pre-ignition.

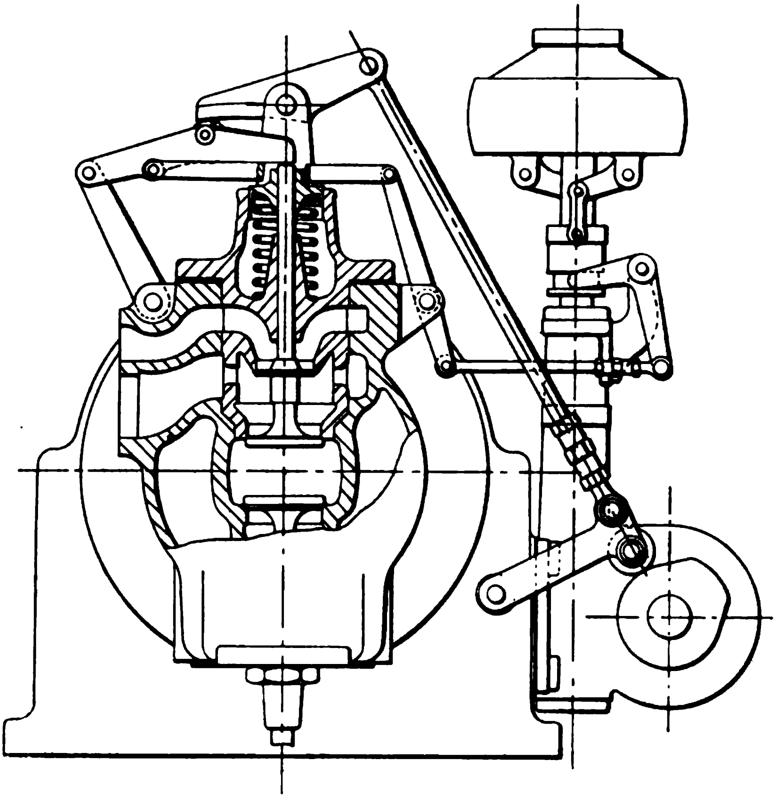


FIG. 67

It is obviously better to have a safe, smoothly running engine which falls short of the best theoretical diagram than to have an engine producing the most admirable thermal results but always on the verge of danger from pre-ignition.

Increase of combustion chamber surface in cylinders above 14 ins. diameter produces little increase in loss by cooling, and above that diameter cylinders are now designed purposely to keep up cooling surface. This is very evident on comparing the sections of the larger engines.

Fig. 67 is a transverse section showing the valve and governing arrangement of a recent engine constructed by Messrs. Tangye, Ltd. The inlet and exhaust valves are arranged top and bottom, opening into a port which passes longitudinally into the combustion space. The inlet valve with the gas valve is contained within a sleeve or cage which seats in a bored-out cylindrical passage. The air is admitted to the lower part of the cage by slots and a surrounding annulus, while gas is admitted at the upper part.

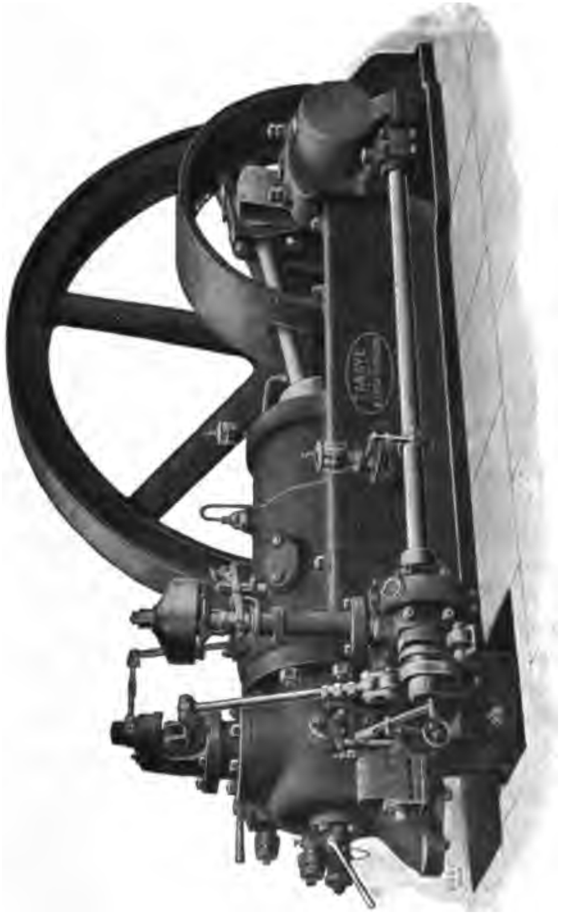


FIG. 68

The gas valve is carried on the inlet spindle, and it seats so as to close off gas from air when the main inlet valve is seated. The governing is on the 'quantity' principle, and both air and gas are throttled for light loads. The arrangement for accomplishing this consists of an upper lever rocked by a cam on the side-shaft. This lever at one end has a curved under-surface over which there is travelled by the governor a pivoted contact piece carried by a horizontal lever pivoted to a vertical one. The governor by a suitable linkage pulls this horizontal lever to and fro, and so varies its leverage that the inlet valve lift is varied and reduced or increased as required by the governor. The arrangement is very similar to that first used on large engines by the Deutz Co., to be described later in the chapter. In the Tangye engine this gear operates very effectively, as is shown by a test made by Mr. Mathot in 1909, the engine working with producer gas from anthracite.

'The engine had a piston of 16½ ins. diameter and a stroke of 23 ins., and ran at a speed of 190 revolutions per minute. The trial at the rated load of 68 BHP was continued for ten hours. Five hours' test was made of the engine at 81 BHP, and an overload test of 88·75 BHP for sixteen minutes.

'The mechanical efficiencies at the various loads were 84·2, 86·3, and 90·2 respectively. The consumption of anthracite amounted to only 0·72 lb. for the rated HP, and 0·665 lb. for the maximum load test, both on the basis of BHP hour, the gross thermal efficiencies based upon the number of B.TH.U. in the coal being 0·248 and 0·269 respectively per BHP. The latter figure is the highest that the author (Mr. Mathot) has had the opportunity to ascertain up to the present, this being due as much to the efficiency of the producer as to the high average pressure and high mechanical efficiency of the engine.'

Messrs. Tangye are to be congratulated on the results obtained, which are only possible with a high compression ratio and a very high efficiency producer.

Fig. 68 is an external view of a similar engine, which shows clearly the various parts, including the magneto for ignition.

Enough has been now described to give some idea of the development of the gas engine in Britain. For the full details of the many excellent engines of British manufacture now on the market the reader is referred to the engineering journals, which now devote ample space to this great and rising competitor with the old-established steam engine and the newer steam turbine.

LARGE GAS ENGINES

The large gas engine movement began in 1895, when the late Mr. B. H. Thwaite proved at the Glasgow Iron Works that blast furnace gas could be commercially applied to operate gas engines.

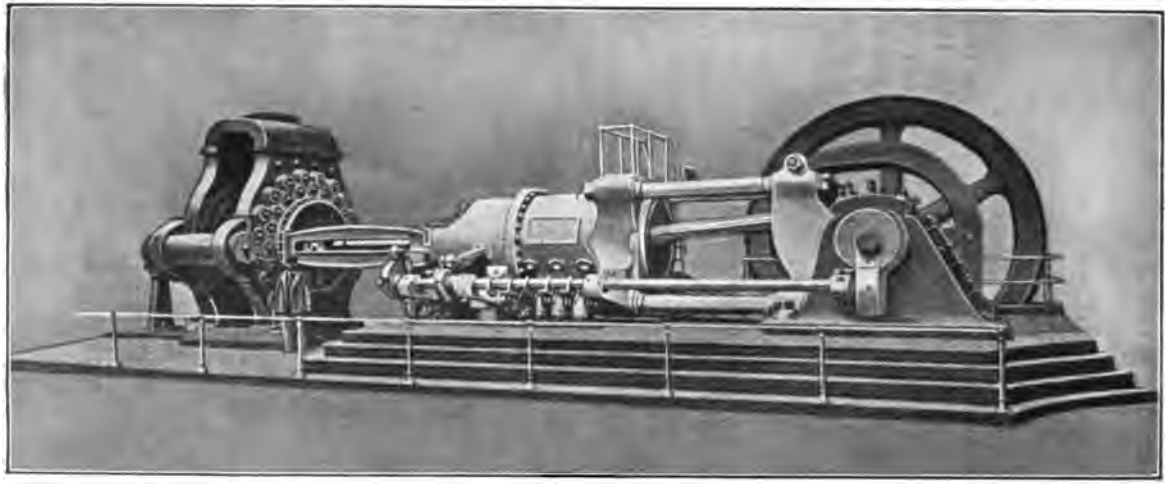


FIG. 69

To the late M. Delamere-Deboutteville and the Société Cockerill belongs the credit of first designing and building engines of large cylinder diameter and great power. Their first large engine was working at Seraing in 1898; its cylinder diameter was 31·5 ins. and stroke 39·5 ins., and it indicated 213 HP at 105 revolutions per minute, with blast furnace gas of 110 B.Th.U. per cub. ft.

In 1899 they surprised the engineering world with a large single cylinder, single acting Otto cycle engine, 51·2 ins. diameter by 55·1 ins. stroke, which gave 600 BHP at 90 revolutions per minute.

Tests made by Professor Hubert of the Liège University in 1900 showed that at its best load it converted 28 per cent. of the heat given to it into indicated work. It was inspected by the author at Seraing in 1901, and it was then performing its work with great regularity and smoothness.

Fig. 69 is an external view of this large engine, and figs. 70 *a*, *b*, and *c* show end elevation at the exhaust, end elevation at the gas and air valves, and a section to larger scale through the charge inlet with separate gas and air valves, respectively.

The engine was a huge one for the power developed; a glance at fig. 69 to note the proportions of the engine to the man standing at the left hand between the engine and the blowing cylinder makes the size very evident. In this engine the compression pressure was 135 lbs. per sq. in. and the explosion pressure usually about 240 lbs. per sq. in. The mean effective pressure ranged from 60 to 65 lbs. per sq. in., and at the normal speed of 90 revolutions per minute the maximum brake power was 725 horse.

In this Cockerill engine an open ended cylinder is connected by four steel rods with main bearings, carrying the crankshaft of 18 ins. diameter. The connecting-rod operates a trunk piston in the manner usual in smaller gas engines, but the piston is water cooled on its end surface, and the water is introduced through the end in a continuous stream, by means of a system of jointed hollow links. The piston carries a hollow piston rod which passes through a stuffing-box carried in the end of the combustion chamber and water-jacket casing. Metallic packing is used. The piston rod is continued and operates a crosshead slide, which guides it, and connects it to the piston rod of a blowing cylinder mounted behind the gas engine cylinder. The valves are driven from the usual side shaft clearly seen in fig. 69, and they are of an interesting type. Both charge inlet and exhaust valves are placed below the cylinder; and admission and exhaust are conducted to and from the cylinder by one passage which branches below forming the charge port and the exhaust port; this necessitates a very long facing and distorted water-jacket casting to include both ports. Outside, the casting measures six feet. Fig. 70 *a* shows an end

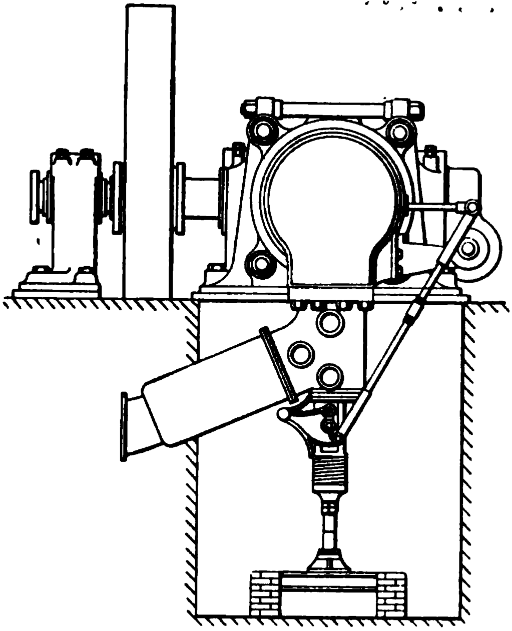


FIG. 70 *a*.

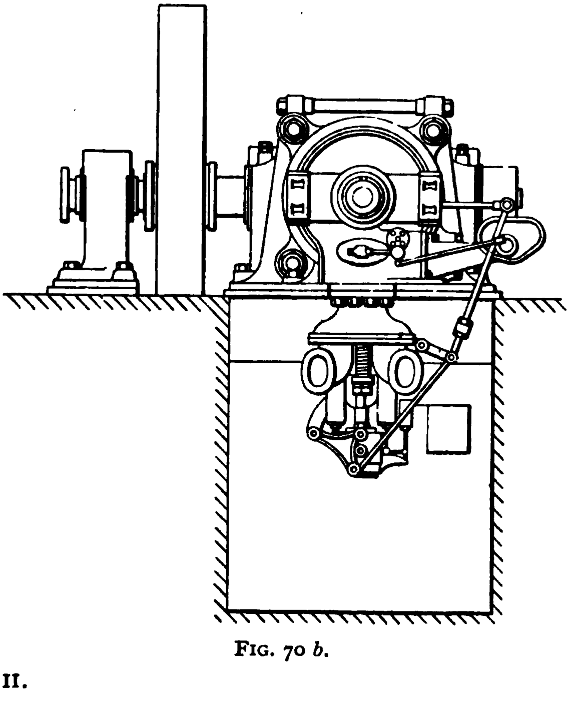


FIG. 70 *b*.

elevation of the exhaust pipe and exhaust valve. The exhaust pipe is water jacketed in order to avoid heating the pit underneath the engine which accommodates the valve.

The exhaust valve is operated by a link from a cam on the side shaft, and the link moves a tumbling lever, seen below. The exhaust valve spring is placed outside the engine, well away from the heat, and the spindle is surrounded by a water-jacketed casing.

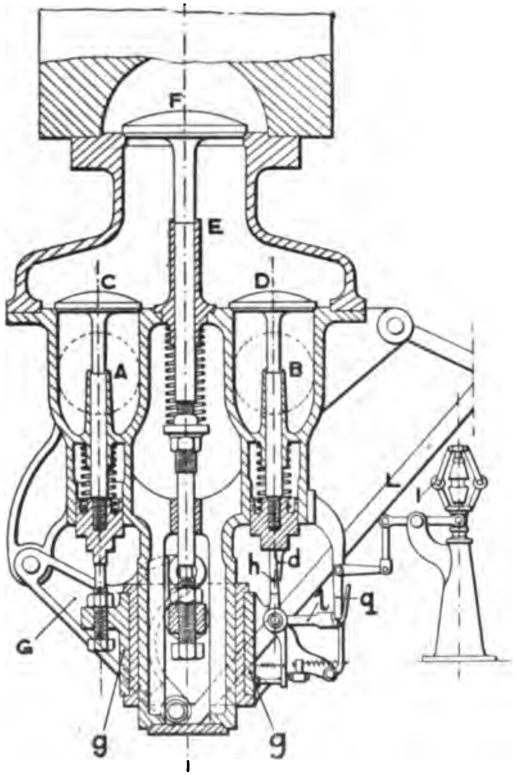


Fig. 70 *c*

Fig. 70 *b* shows an elevation farther back than 70 *a*, and here is seen the gas and air inlet device, operated also from the valve shaft by means of a cam. This figure shows the electric ignition gear actuated by a cam on the same shaft. Fig. 70 *c* shows in section, to larger scale, the arrangement of the inlet, gas, and air valves; F is the inlet valve for the mixed gas and air, C is the air valve, and D the gas valve. The link L is constantly operated by the valve shaft, and it works a tumbler, G, which actuates the sliding sleeve *gg* carrying an adjustable screw on the left-hand side for regulating the lift of the air valve C. On the right-hand side the gas valve D is operated by a hit-or-miss device controlled by the centrifugal governor I. The up-and-down movement of the sleeve *gg* causes a corresponding movement in the attached bell-crank *hl*; normally the end of *l* misses that of the governor-controlled bell-crank arm *q*, in which case the end of *h* engages the projection *d*, and the gas valve is lifted. When, however, the governor rises, the ends of *q* and *l* engage, *l* is depressed, and in consequence the end of *h* misses the projection *d*; the gas valve D then remains closed, and a firing stroke is missed. The engine thus operates in the manner common in smaller and simpler designs, giving either full explosion or no explosion.

The governing is sufficiently good for the purpose of an air blower. The central valve F is opened by the same movement of the tumbler G, and the gas passing through the valve D and air passing by way of the valve C mix in the chamber E, under F, whence the mixture proceeds into the engine cylinder. The gas valve D is also operated from another source under circumstances to be next explained.

In an engine using a water-cooled piston it is absolutely imperative that the water shall always circulate while the engine is running. If the water stops, the piston at once begins to heat up, and it quickly seizes in the cylinder. To prevent this, Messrs. Cockerill's engineers have applied an ingenious contrivance, by which the opening of the gas valve D depends not only on the governor but also on the water supply. The water after passing the piston falls into a small open reservoir, which has a discharge aperture of regulated dimensions. When the water is circulating at a safe rate the level in the reservoir remains constant, the excess supplied being discharged by the aperture. A float in the reservoir is connected by a linkage with the gas valve gear; if from any cause the water circulation stops or becomes insufficient, the aperture lowers the water level in the reservoir, the float falls, and through the action of the connecting linkage the gas valve is not opened and the engine accordingly stops.

The analysis of Seraing blast furnace gas used by the engine is as follows:

ANALYSIS OF BLAST FURNACE GAS AT SERAING

	Per cent.	
Carbonic oxide	27·90	
Hydrogen	1·02	By volume
Marsh gas	7·00	
Carbonic acid	13·95	
Nitrogen	50·10	
	99·97	

The following table summarises the results of an official trial at Seraing by Professors Hubert and Witz:

EXTRACTS FROM RESULTS OF OFFICIAL TRIALS OF 600 HP COCKERILL 'SIMPLEX' ENGINE, MADE AT SERAING, *March*, 1900

ON BRAKE:

Calorific value of the gas, 984 calories per cub. metre = 110·6 B.Th.U. per cub. ft.
Thermal efficiency on IHP with full load, 25·20 per cent.
„ „ „ BHP „ „ „ 20·48 per cent.

COUPLED TO BLOWING CYLINDER:

Calorific value of the gas: 1st series, 991 cals. = 111·3 B.Th.U.
2nd series, 1,004 cals. = 112·8 B.Th.U.
Thermal efficiency on IHP: 1st series, 27·34 per cent.
2nd series, 27·11 per cent.
Thermal efficiency on compressed air: 1st series, 20·40 per cent.
2nd series, 22·17 per cent.

—	Admissions per min.	IHP	Revs. per min.	HP in compressed air	Pressure mm. Hg	Mechanical efficiency of system	Vol. of gas consumed per min. at 0° C. and 760 mm. of mercury	Consumption of gas per hour in cub. metres per	
								IHP	Effective HP
						Per cent.	c.m.		
First series .	36·30	746·21	84	562·55	394	75·41	29·190	2·337	3·115
Second series	46·51	886·48	93	725·57	450	81·81	34·487	2·334	2·854

The motor run light absorbed 147·36 HP.

Professor Hubert gives the following approximate B/S of heat quantities:
Heat converted into work in cylinder, 28 per cent.
Heat carried away by circulating water, 52 „ „
Heat carried away by gases and losses, 20 „ „

The gas consumption was thus at best 3·1 cub. metres per BHP hour and 2·33 cub. metres per IHP hour. The efficiency figures given above are calculated on the lower heating value of the gas.

Many of these engines are installed on the Continent, but the first used in England was erected at the Ormesby Ironworks, Middlesbrough, of Messrs. Cochrane in 1901, where it was inspected by the author. A year's experience of this engine was described by Mr. Cecil A. Cochrane at a meeting of the Cleveland Institution of Engineers in 1902.[1]

Mr. Cochrane spoke in favourable terms of the working of the engine, and gave the following particulars:

---

[1] 'The Use of Blast Furnace Gas in Gas Engines,' Cleveland Institution of Engineers, Dec. 9, 1902.

'The 600 HP engine in operation at the Ormesby Ironworks was built by the Société John Cockerill, of Seraing, Belgium, and is of the single cylinder "Simplex" type as patented by M. Delamere-Deboutteville, and is direct coupled to a double-acting blowing cylinder. The principal dimensions are:

'Gas cylinder 1·30 m. or 4 ft. 3 ins. diameter.

'Blowing cylinder 1·7 m. or 5 ft. 7 ins. diameter.

'Stroke 1·4 m. or 4 ft. 7 ins.

'The normal speed of running is 78 revolutions per minute, the blast being delivered at a pressure of 7 lbs. per sq. in. The capacity of the engine is 500 cub. metres (= 17,657 cub. ft.) per min. at 40 cm. Hg (= 7·731 lbs.) or 310 cub. metres (= 10,948 cub. ft.) per min. at 76 cm. Hg (= 14·7 lbs.).

'The flywheel is in halves and weighs 33 tons, the diameter being 16 ft. 6 ins., while the whole engine weighs 160 tons and occupies a space 52 ft. 6 ins. by 21 ft. 9 ins. It is a similar engine in all respects to that exhibited by the Cockerill Company at the Paris Exhibition. The system is the ordinary Otto cycle, giving one impulse every other revolution. Governing is effected on the hit and miss principle, and by means of the heavy flywheel remarkable regularity is obtained.

'It is claimed by some makers that governing by throttling the supply of gas is the most satisfactory method, inasmuch as greater regularity in running is thereby obtainable. It is, however, more wasteful in the consumption of gas than the hit and miss method, and certainly no greater regularity is required in a blast engine than that obtained.

'The charge is fired electrically. This is effected by means of a small slide valve, automatically worked by the engine itself, travelling between two slide faces, in each of which there is a small opening, these openings being of equal size and in line with one another. The opening in the lower valve face allows direct communication to the combustion chamber and cylinder. There is also a similar opening in the slide, and at a certain point of the travel of the valve the three openings coincide. Electrical contacts are placed in the openings of the moving slide and the upper valve face opening, either of which can fire the charge when the three openings begin to coincide. The slide valve is of course so set that the firing takes place at the required moment, and this can be regulated while the engine is running between limits represented by 10 mm. of the travel of the slide. The adjustment is made possible by the employment of a graduated disc having ten notches, and to which the slide valve is attached, each notch representing 1 mm. of valve travel. No trouble has been experienced up to the present time with the ignition of the charge.

'There is no special arrangement for scavenging in this engine. The combustion chamber and cylinder are however swept out when the governing apparatus comes into operation, a charge of air only being drawn in and expelled. As a consequence, the first explosion after a 'miss' is always of the greatest intensity, subsequent explosions giving a reduced initial pressure owing to the fouling of the air in the combustion chamber by the products of combustion. The number of misses varies from 1 in 4 to 1 in 30, according to the quality of the gas and the pressure against which the engine is blowing.

'The cylinder, exhaust valve, and main bearings are water jacketed, and cooling is so complete that even after the engine has been running several days the cylinder walls are cold to the touch. The water for this cooling is derived from a tank which forms the roof of one of our engine houses, whence it is conveyed to and passes through the engine with a head of about 60 feet. After doing its work, it is discharged into a small tank fitted with a float, so that if by any chance the circulating water fails, the float sinks and automatically stops the engine by putting the governing apparatus out of gear. From this tank the water passes into a reservoir, whence it is pumped by a compound pump (with 8 in. and 12 in. steam cylinders, and 10½ in. plungers) to the top of a Klein open type water cooler, which is fixed in the tank aforementioned above the engine-house.

'The piston is also cooled, but in this case it is necessary the water should be supplied at a pressure of about 50 lbs. per sq. in. to overcome the reciprocating action of the piston when the engine is running.

'The head of water at our disposal not being sufficient to give this pressure, we employ a small subsidiary pump (with steam cylinders 5½ ins., and plungers 4¾ ins. diameter) for this purpose.

'At first, in order to avoid any possibility of the piston heating, we used waterworks water fresh from the main, but we have since found that the ordinary jacket circulating water will suffice, thereby effecting a considerable saving.

'The quantity of water required for cooling the cylinder and piston was furnished to us by the makers as 32 and 8 cub. metres per hour respectively, or a total of 40 cub. metres per hour = 8,800 gallons.

'We have made no effort to economise water for cooling purposes, having merely to pump it round and round the engine.

'The quantity of water passed round our engine is 25,398 gallons per hour, and the rise in temperature sustained by the water passing through the jacketing has been several times determined and found to be only 6° F.

The starting of the engine is effected by drawing into the cylinder a charge of naphtha vapour through a specially designed carburetter, which is fitted to the engine. This charge is then slightly compressed by reversing the flywheel and exploded by electrical ignition.

'The impulse given to the engine is sufficient to cause it to make two revolutions, during the second of which a charge of gas and air is drawn into the cylinder and compressed, and if the mixture is of the proper quality, it is in turn exploded, and the engine is started. The flywheel is rotated to draw in the initial naphtha charge and compress it by an electric motor of 6 HP, which is fitted with a pinion, which works a spur gearing on the inside of the flywheel. . . .'

It will be noted that the cooling water in circulation is very great, but little appears to be lost, as Mr. Cochrane stated in discussion that only 250 gallons per day is required to be added to make up for evaporation and other losses.

The consumption of lubricant is given as follows:

'With regard to lubrication, I think there is a very general idea abroad that this is a costly matter, and one of the drawbacks of these engines. I have been careful to obtain therefore reliable figures as to the cost of lubrication of our engine for your guidance.

'The cylinder is lubricated by means of six automatic ratchet-ram lubricators, worked by small eccentrics off the valve shaft, the main bearings and connecting rod by ordinary sight-feed lubricators, and the gearing and valve shaft bearings by Stauffer "tell-tale" lubricators. The lubricants used are "Mazout," a Russian mineral oil, which we import in lots of 2,000 gallons at a time, and Stauffer lubricant. During the months of September and October, when the engine ran continuously, except for short stoppages for cleaning the fan, the average consumption of lubricants per day was as follows:

	s.	*d.*
'Mazout,' 8·27 gallons at 7·2*d*. per gal.	5	0
Stauffer, 48 lbs. per month = per day	0	7¾
Total per day .	5	7¾

'This includes the oil used on the blowing cylinder, which is also supplied by a ram lubricator.

'During the year the longest period of continuous running was 15 days.

'In normal running the charge compression was 110 lbs. per sq. in. above atmosphere, the initial pressure of explosion 260 lbs. per sq. in., and the mean effective about 77 lbs. per sq. in. at 80 revolutions per min.'

A trial made at the Ormesby Works gave the following results:

RESULT OF TRIAL OF 600 HP COCKERILL 'SIMPLEX' GAS ENGINE, MADE AT ORMESBY, *December*, 1902

ENGINE COUPLED TO BLOWING CYLINDER:

Calorific value of gas, calculated from analysis, 867·6 cals. per cub. metre, or 98 B.Th.U. per cub. ft.

Thermal efficiency of engine on IHP with full load = 25·43 per cent.
„ „ „ „ compressed air = 19·32 per cent.

Admissions per min.	IHP	Revs. per min.	HP in compressed air	Air pressure in lbs. per sq. in.	Mechanical efficiency of machine	Volume of gas consumed per min. cub. metres	Consumption of gas per hour in metres per	
							IHP	Effective HP
35·5	742	76	564·3	7·00	76 %	38·44	3·108	4·01

Total gas consumed = 2306·4 cub. metres per hour.

The analysis of the gas was:

57·05 per cent. N
26·66 „ „ CO
16·25 „ „ CO_2
0·084 „ „ H
By weight.

The gas was purified from dust by means of two fans supplied from the centre with regulated amounts of water, and in the latter part of 1902 the gas contained as little as 0·030 gram per cubic metre.

The total power consumed by the fans was 26 HP. The pressure of the gas at the engine was 10 ins. water. The limit specified by Messrs. Cockerill for dust was 0·25 gram per cubic metre.

Messrs. Richardson, Westgarth & Co., Ltd., took up the manufacture of these engines in 1901, and built seven 600 HP gas blowing engines for the Cargo Fleet Ironworks Co.; but they found it necessary to modify the Cockerill design in several particulars, as they experienced some trouble with cracked breech ends. They have kindly supplied the author with drawings from which figs. 71 and 72 are prepared.

Fig. 71 shows the breech end of one of these large 51 ins. cylinder engines, as designed on the Continent, and the positions in which cracks developed are indicated by hatched lines. To overcome these serious fractures it was found necessary to re-design the parts as shown in fig. 72, which shall now be compared with fig. 71.

Referring to fig. 71, it will be noted that both charge inlet and exhaust valves are carried on facings below the breech end; A is the

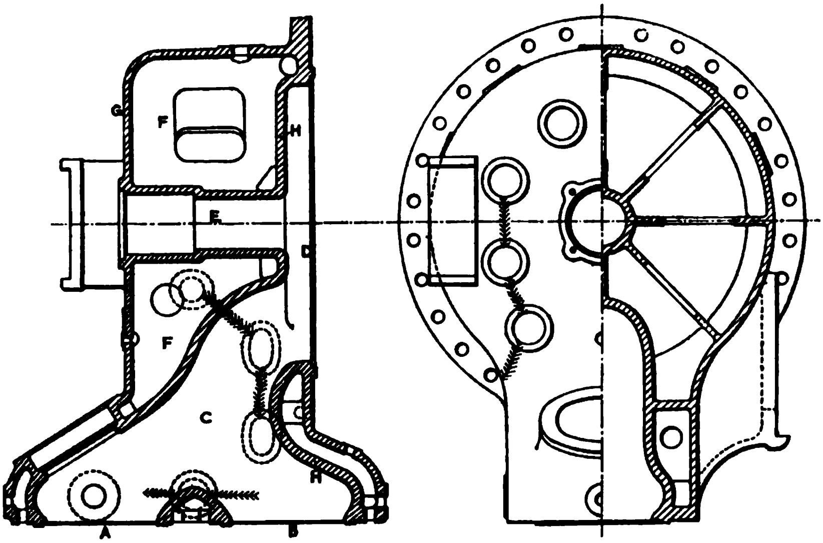


FIG. 71

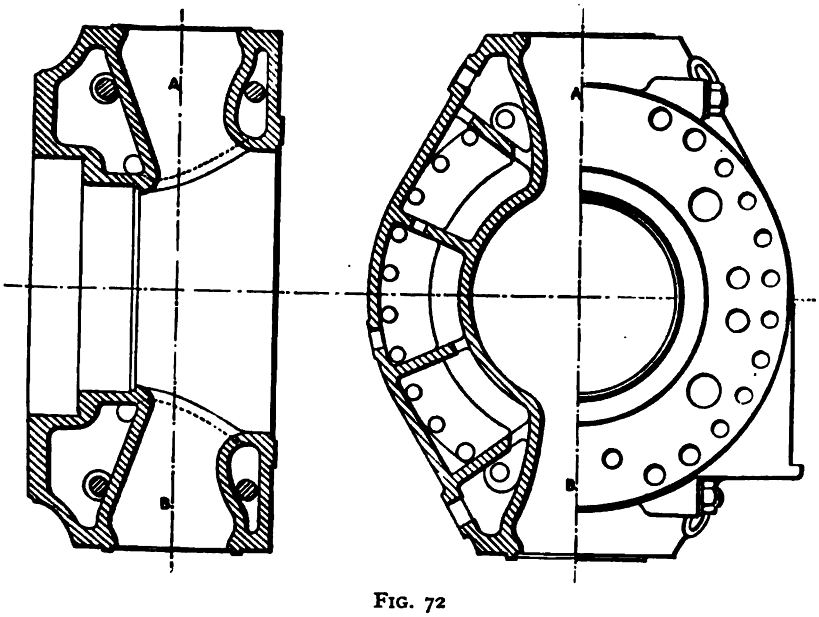


FIG. 72

charge inlet port, B the exhaust port, and both join into a common connecting passage, C, by which the charge is admitted and the exhaust discharged from the cylinder D. The piston rod for operating the blower piston passes through the stuffing-box chamber E. The water jacket F is formed by the casing G and the inner walls H of the breech end. The whole breech end is cast in one piece, and it is obviously of a very unsymmetrical design. The extreme height from the valve flange below to the top of the breech and main flange is nearly 7 ft. 10 ins.; the depth of the casting from back to front is 3 ft. on the axis of the stuffing-box, and the depth from back to front of the casing outside the forked inlet and exhaust passages is 6 ft. Under the circumstances it is not surprising that casting and expansion stresses are set up which frequently produce cracks in the casing in the positions indicated. These cracks start between the apertures in the casing provided for cleaning purposes. The cracks also sometimes pass through the passage E.

The breech end shown in fig. 72 was designed by Messrs. Richardson, Westgarth & Co. to overcome this difficulty, and it was found to be very effective. The inlet and exhaust passages were arranged respectively at the top and bottom, the casting was arranged in ring-shaped form with a central circular opening into which was fitted a separate cylinder cover carrying a stuffing-box and separately supplied with water.

The casting thus became symmetrical, and the central opening for the cover kept down casting stresses to a minimum. The long irregular valve chamber containing the ports A and B was dispensed with, and the breech end was found a great improvement on the original.

Although it was thought advisable to change the design, yet many of the old type worked satisfactorily for years; in June 1908 Mr. John Westgarth wrote to the author: 'It may be useful to know that quite a number of old-type breeches are still running after more than three years' continuous work.'

Although many of these large cylinder engines worked well and gave little trouble, yet it was found that in others great difficulties were encountered due to the cracking of both cylinders and pistons; accordingly many engineers have preferred to keep down cylinder dimensions by using multiple cylinders.

DEUTZ ENGINES

An early large engine of Deutz design was at work in Hoerde in 1901, when the author inspected it. The power was 1000 brake horse, and it had four cylinders arranged *vis à vis* on two cranks similar to the Crossley engine shown at fig. 33.

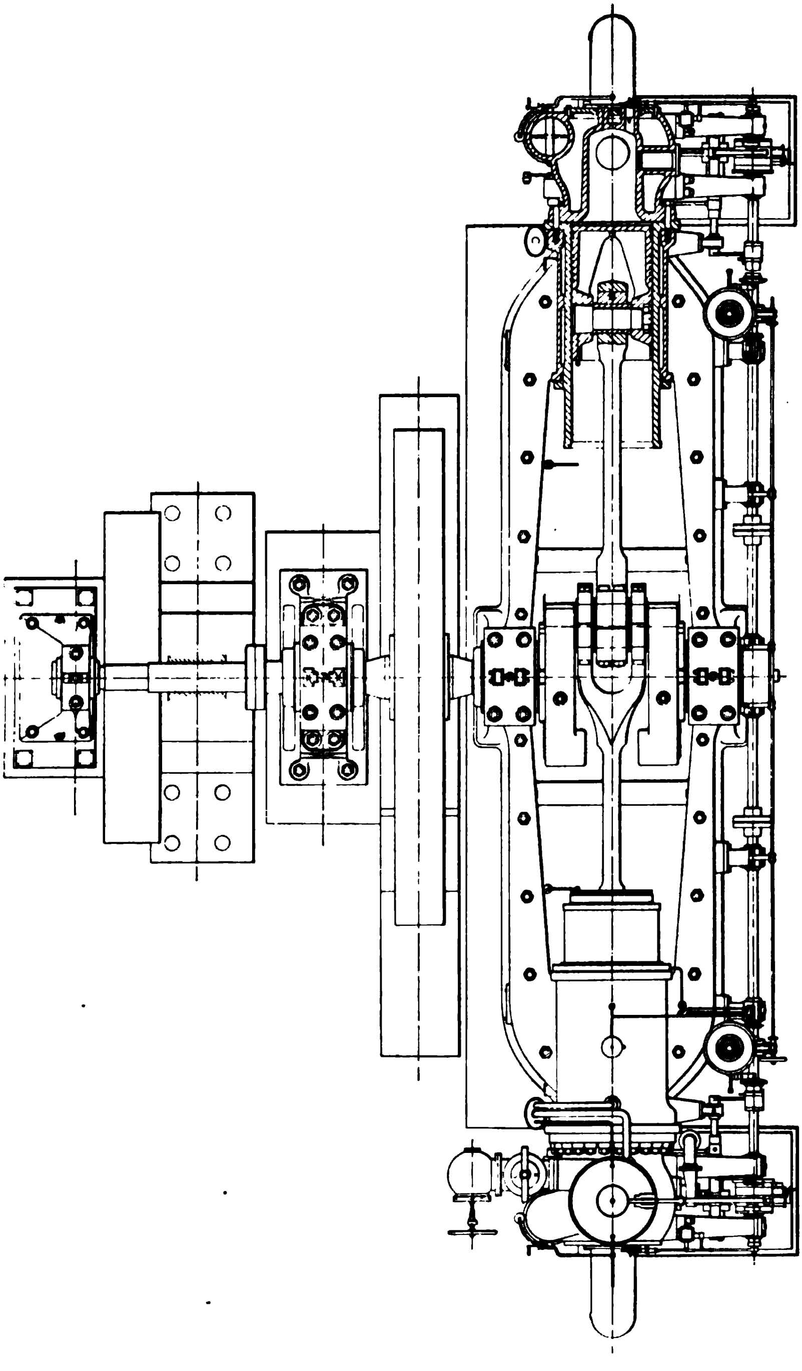


Fig. 73

Fig. 73 shows half the engine in plan, part section; it was then hoped by the Deutz Co. and others to avoid the use of water-cooled pistons, and this engine at first had no water cooling either in pistons or exhaust valves.

The pistons were 33 ins. diameter and stroke 39·3 ins. The diameter of the crankshaft was 16·5 ins.; speed of engine, 135 revolutions per minute, and it gave a maximum of 1200 brake horse-power.

The pistons worked in open trunk cylinders, and were fitted with four white metal rings, a pair on each side of the crosshead pin. These rings took the side thrust due to the connecting-rod and diminished the friction, at the same time taking the wear of the guiding part and saving the liner.

The cylinder liner was carried within the water jacket casing, and was bolted to a facing at its outer end, which facing also carried the combustion chamber or breech end. The front end of the liner passed through packing rings in the water casing, and was thus free to expand as in the smaller engines already described.

The combustion chamber carried above the charge admission valve, and below the exhaust valve.

The inlet valve casing had two annular chambers divided from each other by a perforated plate. The upper division connected to the gas supply, and the lower to air. During the suction period gas flowed from the upper chamber, through the perforations, into the lower chamber, where it mixed with the incoming air. This mixture of gas and air then passed through perforations in the sides of the valve chamber itself, and thus a double mixing was obtained. The valve spindle sleeves were water jacketed.

A throttle valve shown in the plan was used for the purpose of setting the proportion of air.

The engine was started by compressed air.

The pistons were found to overheat and ultimately crack, so that after a time watered pistons were substituted. The largest pistons run without watering in England are found in the Crossley engine of 26 ins. cylinder diameter described in Mr. Humphrey's test, pp. 43 et seq. The largest unwatered pistons used by the National Gas Engine Co., Ltd., are 24 ins. diameter, so that it is not surprising to find 33 ins. diameter pistons requiring water jacketing. It is interesting to note, however, that at the date of the author's visit to Hoerde in 1901, the piston had been at work without water for almost two years; at that date the engines were stopped for the purpose of fitting them with water-cooled pistons. Two of the pistons which had been at work without water cooling were examined by the author, and found to be in a very good condition.

The piston ends had obviously been very hot, judging from the red colour of the metal there; probably they were hot enough to give rise to some pre-ignitions. It is interesting, however, to learn from this experience that pistons of so large a diameter as 33 ins. *can* be run without water cooling.

Electric ignition was used, the device belonging to the type in which a magneto armature is liberated, and operated by a spring; when at the moment of maximum velocity, contacts within the cylinder are separated and a dense low-tension spark is produced.

Governing was effected by varying the point of opening of the gas supply valve. The air admitted was thus nearly constant in volume, while the gas varied.

The mixture at the ignition point within the long combustion space was kept of nearly constant composition, although at the piston end of the combustion space it was more dilute—too dilute to fire without the stronger charge at the ignition end of the combustion space or port.

The long flat port forming the combustion chamber secured that the charge next the electric spark was ignitible under all conditions of load.

The engine governing thus, gives an impulse for every two revolutions in each cylinder.

The following is an analysis of the blast furnace gas used at the Hoerde works:

ANALYSIS OF BLAST FURNACE GAS AT HOERDE

	Per cent.	
Carbonic oxide	32·0	By volume
Hydrogen	2·5	
Carbonic acid	8·5	
Nitrogen	57·0	
	100·0	

The difficulties of the open trunk engine with large cylinders led Continental engineers to abandon that type for large power engines, and they introduced a four-cycle, double-acting engine which enabled them to double the power obtainable per cylinder for a given diameter. Among those who discarded the open trunk at an early date we find the Deutz Gas Motoren-Fabrik, who produced a double-acting engine early in 1903.

Fig. 74 shows an external view, and fig. 75 longitudinal and transverse sections, of an early Deutz double-acting engine having a single cylinder: the rated power is 250 brake horse; cylinder diameter 21½ ins.; stroke 27½ ins.; revolutions 150 per minute.

The general design, it will be noted, follows that of the steam engine; indeed it might be, so far as appearance is concerned, a horizontal steam engine with external valves.

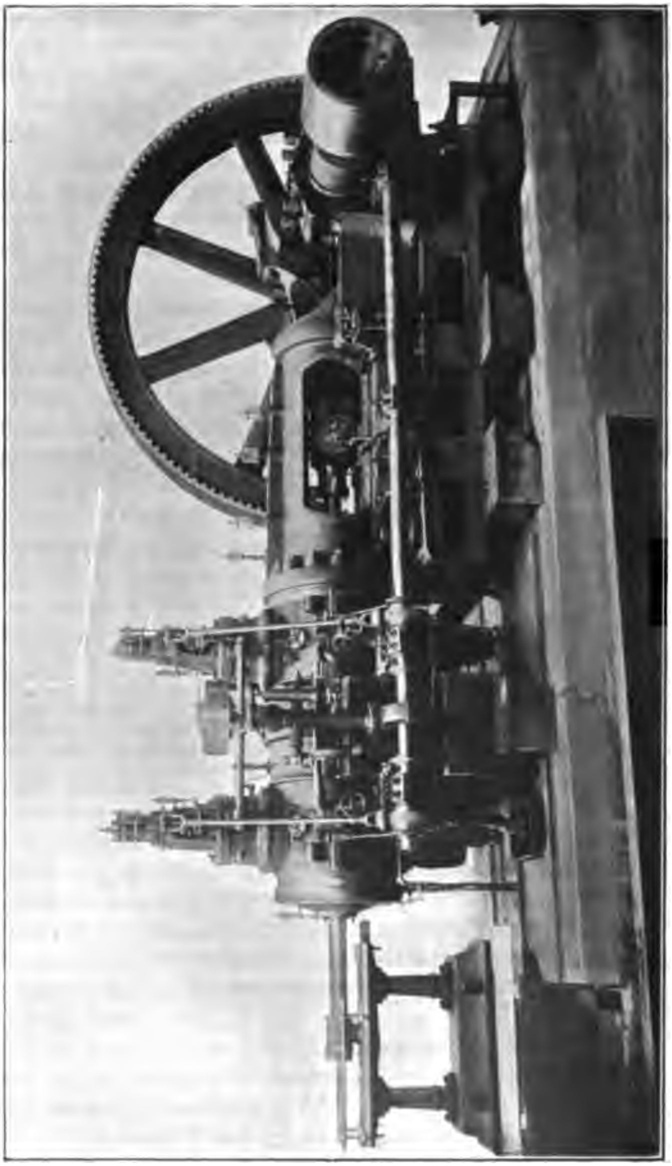


FIG. 74

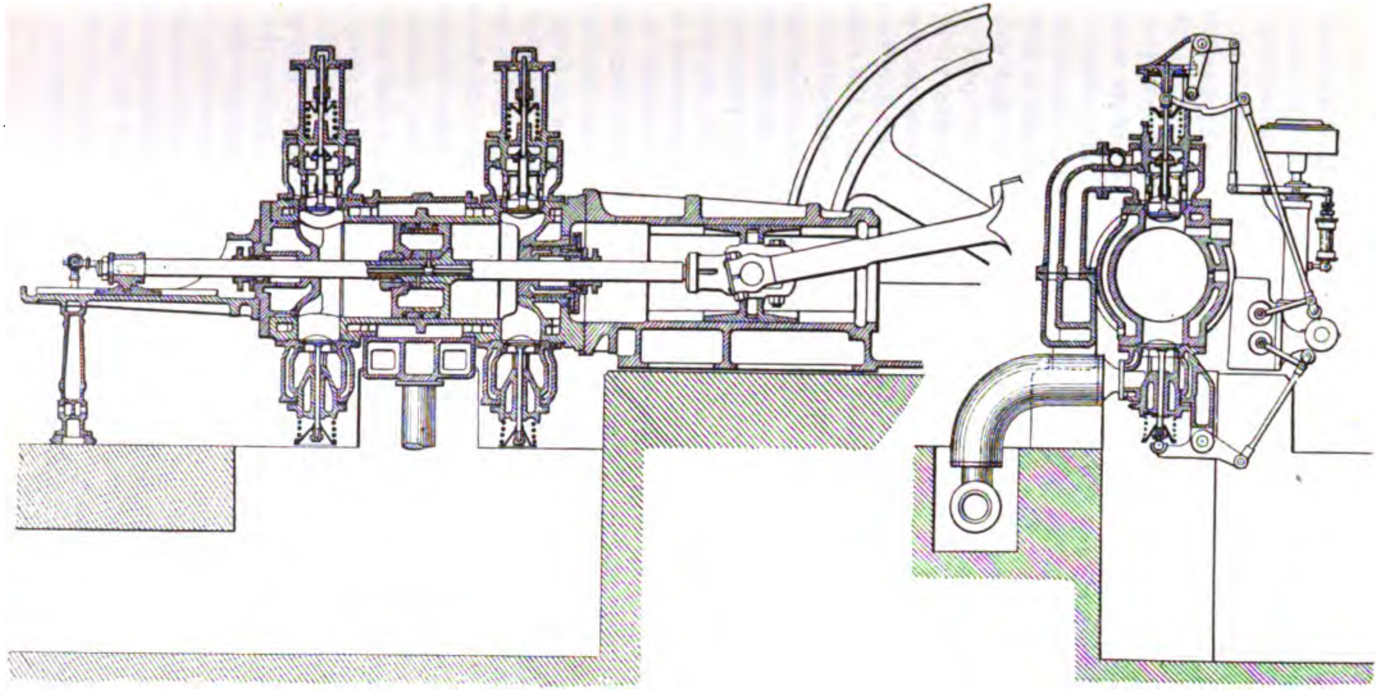


FIG. 75

The engine bed consists of a casting which carries the main bearings and a cylindrical crosshead guide, which terminates in a strong circular flange. To the turned face of this flange is bolted concentrically a cast-iron cylinder, having a water-jacketed front cover cast on, and bored out to take a water-jacketed stuffing-box. At the back end of the cylinder is fitted a cylinder cover, water jacketed and carrying a stuffing-box. The engine so far as described consists of three castings—the bed, the cylinder with its front cover cast on and its valve ports arranged top and bottom, and the water-jacketed rear cylinder cover. The front and rear portions of the casing to form the cylinder water jacket are cast with the cylinder, and reach backward and forward sufficiently to extend in both directions beyond the charge inlet and exhaust ports where the casing is turned externally. The centre part of the water jacket is formed by a cast-iron central bed made in halves and bolted together to surround the cylinder and form the jacket. The water joint is made by packing rings which allow a limited freedom of movement longitudinally and radially; this permits the cylinder to expand free from constraint by the jacket. The central cast-iron bed also serves to support the weight of the cylinder and piston, and for this purpose it is carried on a separate pedestal from the foundation. Otherwise the cylinder would be overhung from the main bed or frame flange. This centre bed also serves to conduct gas and air to the engine by means of chambers formed within it connected to the inlet valves by pipes. These pipes are divided longitudinally, so that channels are formed leading gas and air separately to the inlet valves, so that no combustible mixture is formed until the valves are reached. The piston rod passes right through both stuffing-boxes at front and back; it connects to a crosshead and connecting rod at the front and to a slipper guide at the back. The weight of the piston is thus carried by front and back slides, so that it floats clear of the cylinder sides, and the piston rings alone rub upon the walls to make the piston pressure tight. The inlet valves are placed above the cylinder and the exhaust valves below; both are operated by cams and lever and links from a side shaft driven from the crankshaft by skew gear. In this engine neither exhaust nor inlet valves are watered. The inlet valve sleeve carrying the valve seat, valve spindle, and guides passes into an external casing bolted to a facing surrounding the inlet port. This casing is divided into an upper and lower annular chamber. The sleeve when in position opens by ports into corresponding upper and lower chambers—the upper chamber for gas, and the lower for air. The main valve spindle carries above the main valve a gas valve, which is spring-seated, so that when the main is closed the gas valve is also closed. The lower part of the spindle also carries a cylindrical valve with ports which close and open the air supply chamber.

When the inlet cam operates, the inlet valve rises from its seat and first opens to air alone; increased lift opens the gas valve, and then inflammable mixture passes into the cylinder. The gas valve in the same way shuts slightly before the air valve; this enables the inflammable mixture to be cleared out of the chamber and replaced by air just before the valve closes. By this arrangement a small quantity of air first enters the cylinder on the charging stroke and so prevents back ignition due to the immediate contact of the fresh inflammable charge with the hot exhaust. The gas and air pipes are led under the centre bed, and divided pipes lead separately to the external casing above the inlet valves.

The valve sleeves and valves can thus be removed for cleaning without disturbing the pipe connections. Governing is effected by diminishing the lift of the inlet valves without disturbing their times of opening and closing. The governor moves a bell-crank lever whose end forms a fulcrum for the opening lever. The governor in fact actuates a shifting fulcrum in such manner that one end of the operating lever becomes shorter and the other end longer, and so although the cam gives a constant lift, the inlet valve is given a variable lift, getting less and less as the engine speed increases. The mixture is thus throttled, and the charge weight introduced into the cylinder diminished to give a reduced impulse. The valve lever at one extreme of its movement comes out of contact with the fulcrum formed by the end of the bell-crank lever, and the governor is then freed from all friction, and can settle the new position of the fulcrum, required by the speed, with rapidity and accuracy.

The exhaust valve sleeves are also surrounded by separate casings, which in this case are water jacketed, which casings are bolted to facings under the exhaust ports. The exhaust valve sleeves themselves are watered at the valve spindle guides. The exhaust valves are not watered. The exhaust valves, like the inlet valves, can be removed for cleaning without disturbing the exhaust pipe connections.

The exhaust pipe bend leaving the engine is water jacketed, till it couples to the main exhaust pipe flange running in a trench.

Ignition is effected by low-tension make and break and magneto.

The piston and piston rod are water cooled; for this purpose a pure water supply is required at a pressure of about 50 lbs. per sq. in. The cylinder, covers, and valve casing are also supplied separately with water at a low pressure—about 10 lbs. per sq. in. is sufficient—and the quantity is separately regulated to keep each part at the desired temperature. A safety device is fitted to interrupt the electric ignition upon any failure of water circulation.

The main bearings are fitted with ring lubricators; the smaller bearings and the guides have sight feed drop lubrication, and the crank pin is supplied by a centrifugal ring. An oil pump supplies oil to the piston rod as well as to the stuffing-boxes.

The Deutz Co. state that the oil consumption for lubrication is from 1 to 1·2 grams per BHP per hour.

The engine is started by compressed air.

The removal of the rear cylinder cover with slipper guide permits the piston to be taken out without dismounting any of the valve gear.

The parts referred to in the description will be readily recognised in figs. 74 and 75.

The following particulars are taken from a test made on March 15, 1904, by Prof. Witz and Mr. Mathot:

TEST OF OTTO-DEUTZ SINGLE-CYLINDER, DOUBLE-ACTING GAS ENGINE.
(*Witz and Mathot*)

Dimensions: *diameter* 21·26 *ins.*; *stroke* 27·56 *ins.*

Mean revolutions per minute during test	150·2
Brake horse-power	222·8
Indicated horse-power	278·5
Mechanical efficiency	80 per cent.
Mean effective pressure, lbs. per sq. in.	74
Consumption of anthracite per BHP hour	0·715 lbs.
Heating value of anthracite	14,600 B.Th.U.
Total heat of anthracite in suction producer per BHP hour	10,600 B.Th.U.
Over all brake thermal efficiency—producer + engine	24·2 per cent.
Consumption of water per BHP hour	7·75 gals.

Mr. Mathot gives fuller particulars of the foregoing and a test of the same engine on the preceding day in his paper on large gas engines read at the Institution of Mechanical Engineers in 1905[1] as follows:

TEST MADE ON A GAS PLANT OF A 4-CYCLE DOUBLE-ACTING ENGINE OF 200 HP AND A SUCTION PRODUCER IN THE WORKS OF THE GASMOTOREN FABRIK DEUTZ, COLOGNE, MARCH 14 AND 15, 1904, BY MESSRS. A. WITZ, R. E. MATHOT, AND DE HERBAIS DE THUN

TABLE AND DATA OF THE TESTS AND FIGURES

Piston diameter $21\frac{1}{4}$ *inches*; *stroke* 27 *ins.*

Diameter of Piston-Rods: *Front* $4\frac{3}{4}$ *ins.*; *rear* $4\frac{5}{16}$ *ins.*

Full Load Tests

	Engine	March 14	March 15
1.	Average number of revolutions per minute	151·29	150·20
2.	Corresponding effective load, BHP	214·22	222·83
3.	Duration of tests, hours	3	10
4.	Average temperature of water after cooling the piston	117·5° F.	—
5.	Average temperature of water after cooling the cylinder and valve seats	135° F.	—
6.	Water consumption for cooling the piston per hour, gallons	386	—
6a.	Water consumption for cylinder and valves, gallons	1003	—

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[1] 'The Growth of Large Gas Engines on the Continent,' by R. E. Mathot, *Proc. I. Mech. E.*, June 21, 1905, p. 619.

	Producer	March 14	March 15
7.	Nature and origin of fuel: anthracite coal, 'Bonne Esperance et Batterie,' Herstal, Belgium.		
8.	Heating value of fuel, B.Th.U.	14650	—
9.	Consumption of fuel per hour (plus 35 lbs. during the night of the 14th inst. for keeping the generator fired during 14 hrs., the engine being stopped), lbs.	199	160
10.	Water consumption per hour in the vapouriser, gallons	—	14
11.	Water consumption per hour in the scrubbers, gallons	—	315
12.	Average temperature of gas at the outlet of the generator	—	558° F.
13.	Average temperature of gas at the outlet of the scrubbers	—	62·5° F.
	Efficiencies		
14.	Gross consumption of coal per BHP hour, lbs.	0·927	0·720
15.	Consumption of coal per BHP hour after deducting the moisture	0·907	0·704
16.	Thermal efficiency relatively to the effective HP and to the dry coal consumed in the boiler, per cent.	19	24·4
17.	Water consumption per brake horse-power hour:		
	For cylinder, stuffing-boxes, valve-seats, and jackets, gallons	4·65	—
	For the piston and piston-rods, gallons	1·75	—
	For the vapouriser, gallons	0·0625	—
	For washing the gas in the scrubbers, gallons	1·42	—
18.	Water converted into steam per lb. of fuel consumed in the generator, gallons	0·0875	—

If it be assumed that the efficiency of the producer is 80 per cent., then the brake efficiency of this engine is $\frac{24{\cdot}2}{0{\cdot}8} = 30{\cdot}25$ per cent. This is a very high brake efficiency for an engine at practical work.

The single cylinder, double-acting four-cycle engine has the disadvantage that on one revolution it receives two power impulses, and on the following revolution the cylinder and piston are wholly occupied in charging and exhausting. Thus although two impulses are given for every two revolutions, these impulses are not given one for each revolution.

By adopting, however, *two* double-acting four-cycle cylinders and arranging them in tandem form, four impulses are obtained for two revolutions. A single crank thus receives two impulses per revolution, exactly like a steam engine.

The advantage of the horizontal tandem double-acting arrangement is so great that practically all Continental engineers now build their large engines of this type.

For the larger engines the Deutz Co. build such a combination,

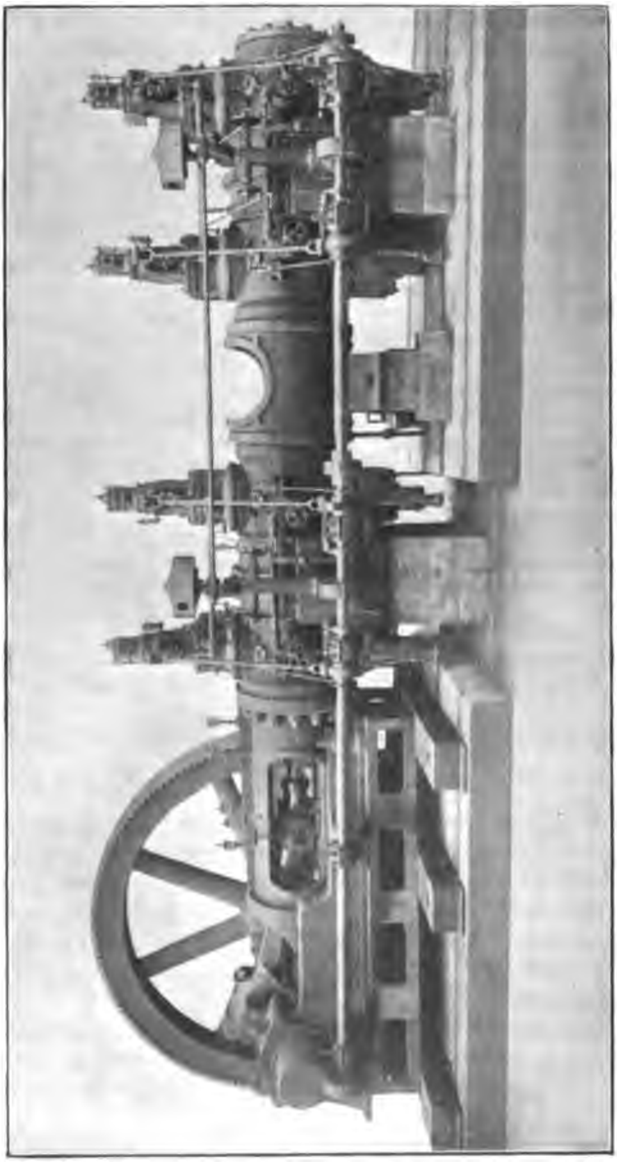

FIG. 76

and fig. 76 is a photograph of one of these large engines, which resembles closely in detail the single cylinder engine just described.

For the larger cylinders, however, a different construction is adopted. Fig. 77 is a section of one of the tandem cylinders of a Deutz 2000 HP engine.

The cylinder diameter is 43·2 ins. and the stroke 51·2 ins. This cylinder consists of a central portion, A, in which the piston works; this is the cylinder proper, and it carries strong flanges at its ends by which cylindrical parts B, B, having cast in them the valve ports, are bolted at each end. The three parts A, B, B form the cylinder proper. Around the open centre is arranged the central bed C, which is divided into two parts and bolted on as a clamp in the same manner as already described for the smaller engine. Hollow cylinder covers are applied at both ends. By this plan the cylinder is built up of four separate castings, and has two cover castings applied at its ends. This plan of casting in separate pieces and building up is also used by other Continental engine builders, but many still prefer to cast cylinder and jacket with all parts in one piece.

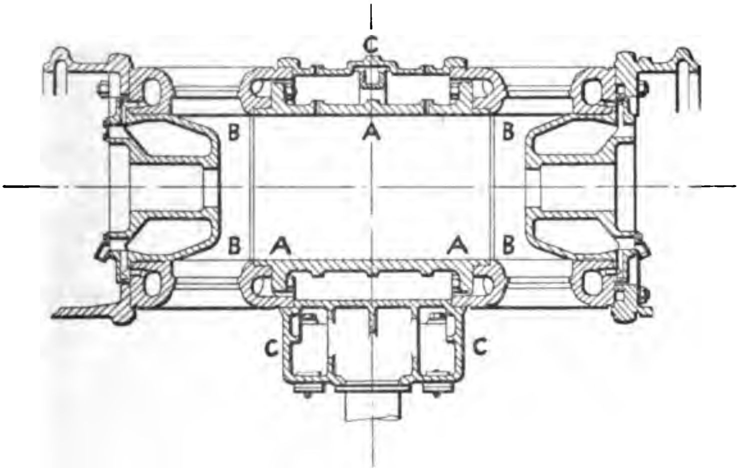

FIG. 77

Even the Deutz Co. appear to prefer this casting in one piece, under some circumstances, for a late cylinder casting so made is shown at figs. 78, 79, and 80.

These figures also show another arrangement of valves used by this company for large engines. It will be seen from fig. 78 that the whole cylinder with its water-jacket casing is cast in one piece and cored out for charge admission and exhaust ports, and also provided with ports for compressed air starting and electric ignition. The compressed air starting valve and the electric ignition plugs are shown clearly in fig. 79, which is a transverse section through the cylinder and water jacket at the plane of the inlet and exhaust valves.

Separate gas and air control valves are placed in the casing which contains the main charge inlet valve sleeve, and the gas and air are

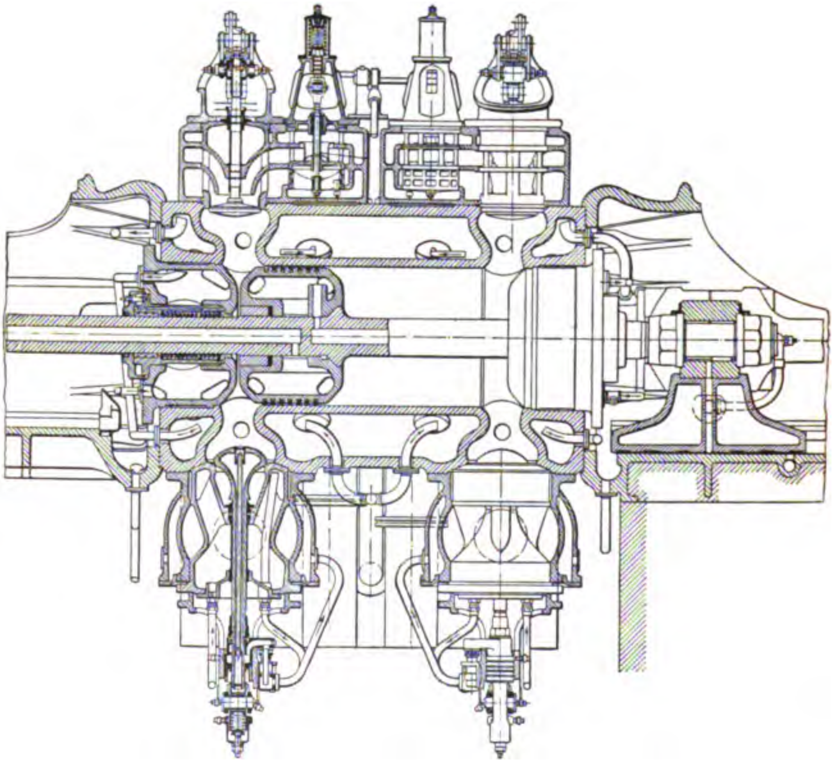

FIG. 78

supplied to separate passages in the casing, so that mixture of gases occurs only at a position just under the inlet valve. No mixing is permitted in a separate chamber, but only at the valve itself. The separate gas and air valves are clearly shown at fig. 78, and in a transverse section at fig. 80.

Governing is effected by the control of these separate gas and air valves, through the action of a cam mounted in an eccentric sleeve; the cam operates a lever connected by a link of the valve spindle. The governor alters the position of the eccentric sleeve, and varies the action of the cam so as to vary the cut off of the gas and air. A gas throttle valve controlled by a handwheel is arranged behind the cam operated gas valve.

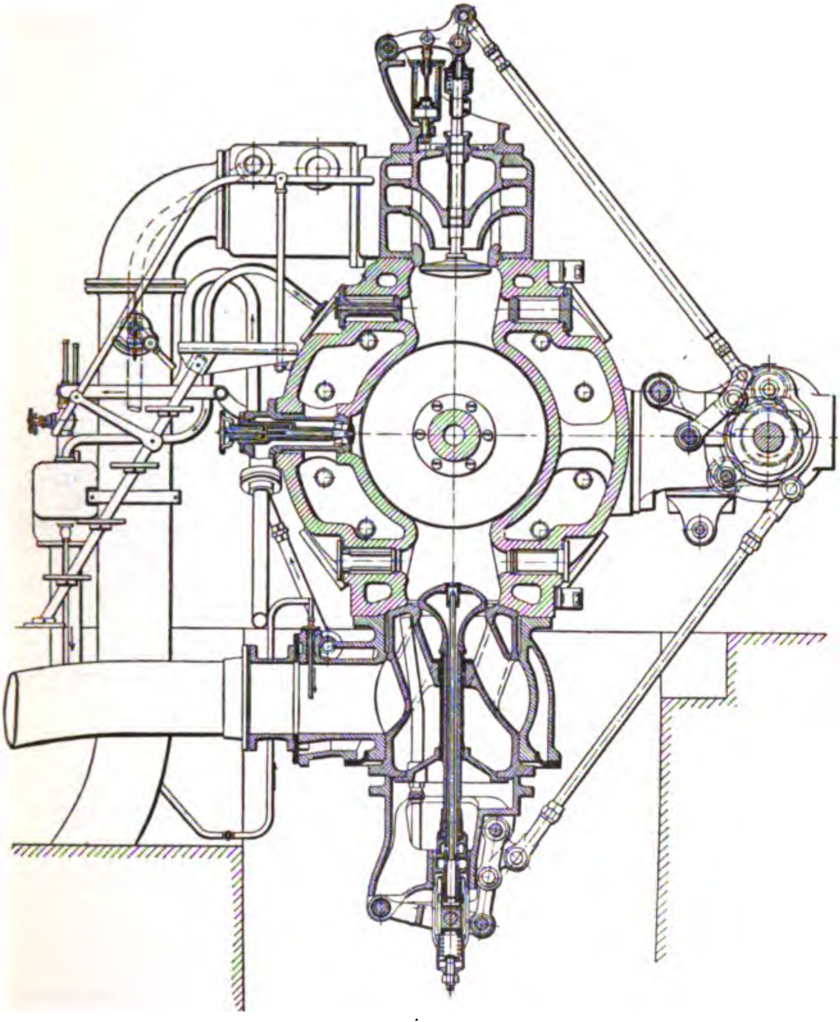

FIG. 79

Both inlet and exhaust valve sleeves fit in casings permanently bolted to facings on the main water casing. Both exhaust valve and its casings are water jacketed. The exhaust valve is operated from a lever and cam on the side shaft, so that the connecting link acts to open against the internal pressure of the exhaust gases while in tension.

The tension is reduced to a moderate amount by the toggle linkage shown at the exhaust spindle end of the link. This toggle arrangement gives a very slow opening movement until the valve is lifted from its seat, followed by a rapid movement after the release of pressure. In this way the pressure of the cam upon its roller is reduced at the time when it is most wanted.

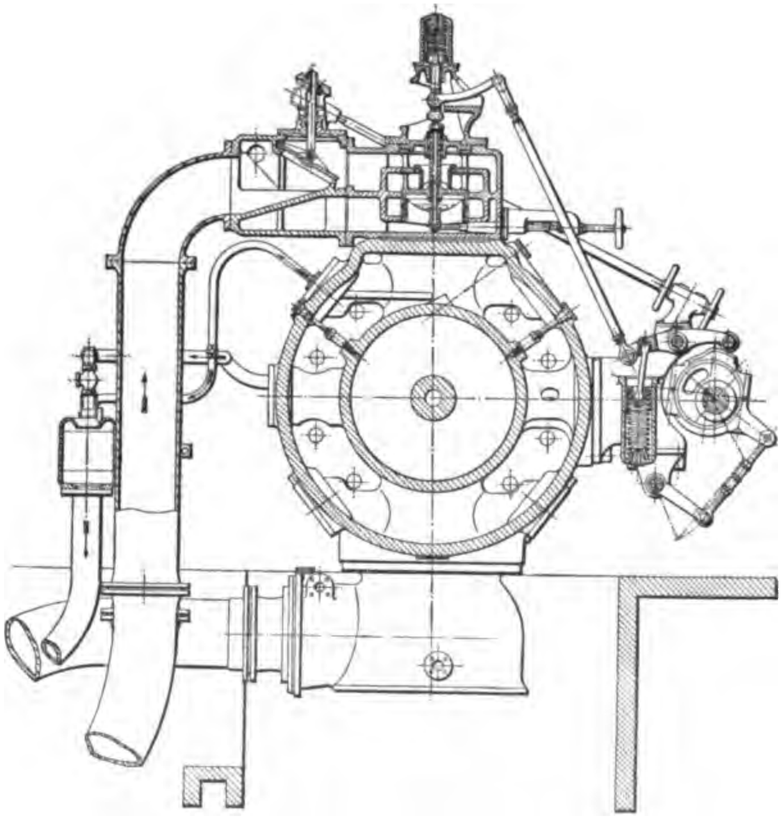

FIG. 80

The charge inlet valve spindle is connected to its operating lever by a spring linkage clearly seen in figs. 78 and 79, so that the spring is compressed for some distance after the closing of the valve; this device secures quiet seating of the valve and allows for wear in the cam, roller, and bearing surfaces of the connecting linkage.

NUREMBERG ENGINES

At first sight an engineer would naturally favour the case of the built-up cylinder for large engines, and experience with free liners in smaller gas engines would suggest the necessity for freedom of expansion between liner and water jacket casing in large diameter engines. A cylinder for a double-acting four-cycle engine, cast with water-jacket casing and all passages and ports, to give 1000 horse is about 9 ft. long by 6 ft. external diameter.

Such a casting would be regarded as formidable in any foundry; casting stresses of unknown amount would appear to be inevitable. Further, allowing only 80° C. temperature difference between the cylinder and water casing walls, we should get while the engine was working a difference of 0·1 in. for 9 ft.

To allow for this something must bend, and engineers prefer to bend cast-iron ends as little as possible.

Notwithstanding these apparent difficulties, German engineers have been successful in building large cylinder engines having cylinders and jacket casing cast in one piece.

One of the most successful firms in this line of work is the Maschinenfabrik Augsburg Nurnberg A. G., who build the engines known as the Nuremberg gas engines.

In June of 1910 these makers state that they had in operation or under construction 444,660 BHP in double acting gas engines.

Fig. 81 shows a general view of a recent double acting tandem Nuremberg gas engine of 2500 BHP.

Fig. 82 is a longitudinal section of a similar engine.

Fig. 83 is a longitudinal section through the cylinder on a larger scale.

Fig. 84 is a transverse section through the valves, showing also the valve operating gear.

The Nuremberg Co. thus describe their engines:

'The Nuremberg gas engine operates on the four-cycle principle, and is double acting, i.e. both ends of the cylinder are closed, and both sides of the piston are arranged for receiving power impulses. As a rule, there are two cylinders arranged in tandem, which work together in such a manner that every stroke has one power impulse. In this manner the well-known thermal advantages of the four-cycle engine are retained, and at the same time all parts of the engine are fully utilised; consequently its dimensions, even with large outputs, are kept within moderate limits, and its turning moment is equal to that of a steam engine with the same number of cranks.

'The frame, cylinders, connecting pieces, and rear guide are registered to each other, and consequently the central position of the

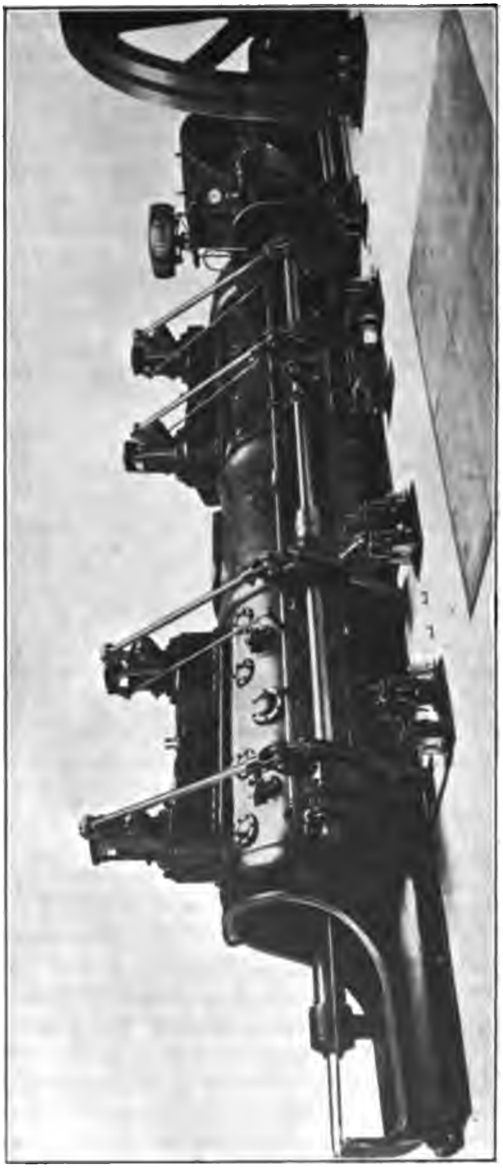

FIG. 81

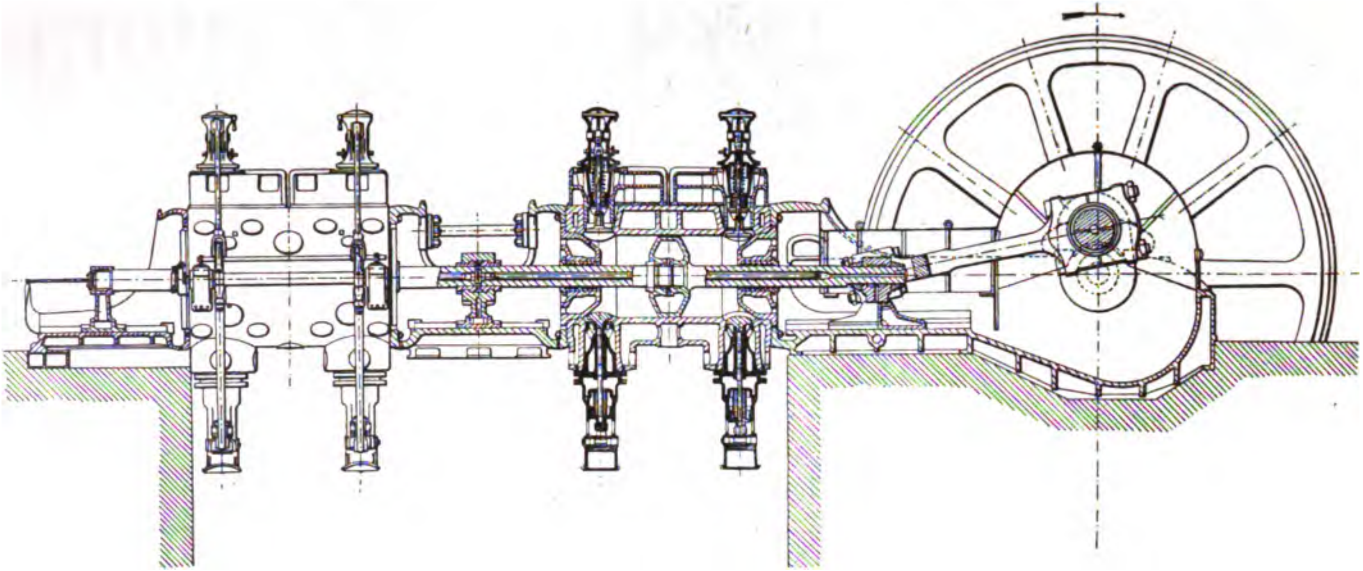

FIG. 82

engine-axis is once for all assured. The frame is rigidly connected to the foundations, whilst the cylinder, guides, and connecting piece are arranged so that they are free to move in a longitudinal direction in order to allow for expansion due to rise of temperature.

'The bed, consisting of a very heavy and massive box casting, rests throughout its entire length upon the foundations, to which it is connected by strong holding-down bolts. In addition it is grouted into the foundations, so that it forms one piece with the foundation block. The bed carries the crosshead guide and the two main bearings for the crankshaft, and at the same time it serves as a receptacle for the oil from the crank. To ensure easy access to the crosshead and cylinder, it is open at the top. The rear end is formed with a complete circular facing or flange of box section, to which the front cylinder is fastened by substantial bolts.

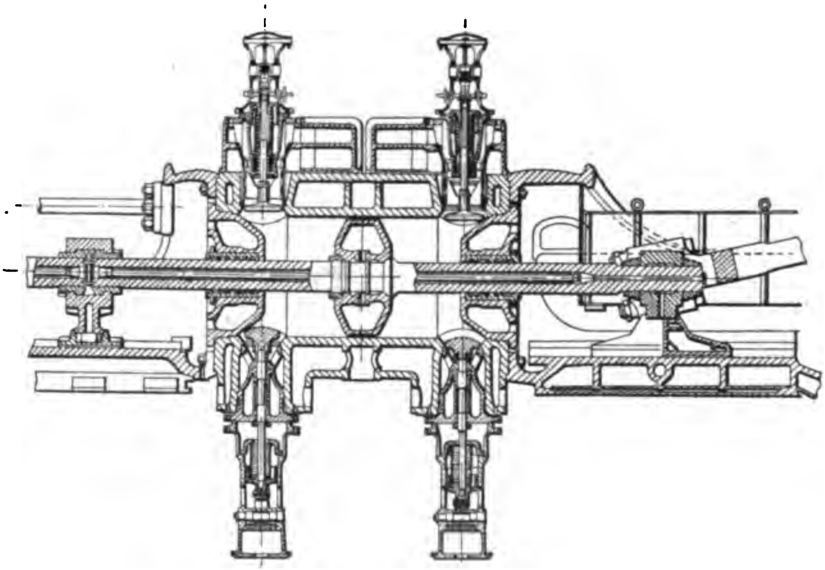

FIG. 83

'In consequence of the symmetrical form of the cast-iron cylinder, unequal stresses in the casting due to the heat are prevented as far as possible. Its inner walls are of regular circular section, and abrupt changes of section are carefully avoided. The spacious water jacket, together with the numerous holes, permit of a ready and thorough cleaning from mud and deposited scale. The water supply to the various parts which particularly require cooling is ample. The broad flange at the end of the cylinders prevents excessive strains occurring on account of the difference in length of the two unequally heated cylinder walls.

'The cast-iron connecting piece is in the form of a tube, strengthened by ribs arranged longitudinally; it connects the two cylinders, and has a cylindrical guide for the piston rod coupling. This connecting piece has a side opening, which is sufficiently large to readily allow of dismantling the cylinder covers and of access to the cylinders.

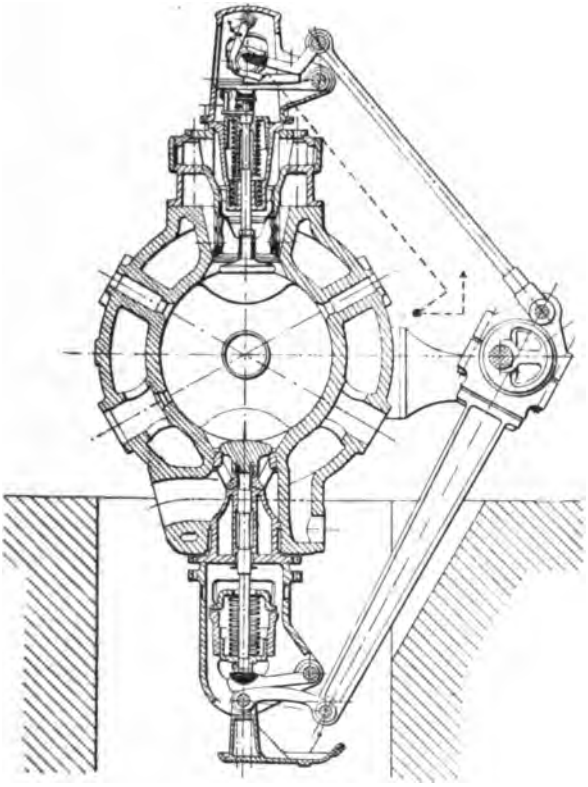

FIG. 84

'To correct the deflection due to their own weight, as well as that of the water-filled piston when supported at the ends only, the hollow piston rods are turned with an upward camber, so that the piston rod, when fastened on the crossheads and loaded by the piston, forms an exact straight line.

'Consequently, the whole weight of the pistons and rods is taken up by the three slides outside the cylinders, and the pistons float in the cylinder, as also do the piston rods in the stuffing-boxes. This arrangement of the pistons not only saves wear and tear of the cylinder and stuffing-boxes to a very great extent, but also considerably decreases the friction of the engine. The pistons are designed as hollow ribless castings, in fact they are substantially-dimensioned disc pistons of simple design, each fitted with six self-tightening packing rings. Each piston can readily be adjusted to the exact centre of its own cylinder, independently of the other and of the possible wear of the moving parts. The front crosshead and the two gudgeon pins are forged from one piece of nickel steel. The piston rod end which passes through the crosshead is threaded on both sides of the portion fitting in the crosshead, and is firmly secured by nuts and lock-nuts. The crosshead slippers are lined with white metal. The connecting-rod is made of Siemens-Martin steel, the crank end is of the marine type, and the rear end of the fork type. The crankshaft is either forged in one piece of Siemens-Martin steel, or is built up in several pieces. It is supported at either side of the crank by brasses in four parts, which are lined with white metal. These brasses can be evenly adjusted vertically as well as horizontally. Forced lubrication, through the hollow shaft and from the crosshead slide, is provided for the journals.

'The piston-rod stuffing-boxes consist of a number of cast-iron and white metal rings of similar section, made in three parts. These rings are pressed against the rods by means of spiral springs, which are evenly distributed on their circumference.

'Lubricating oil, under pressure, is led to the middle of the stuffing-boxes. This type of stuffing-box ensures that all wear is confined to the packing-rings, which can readily be adjusted or renewed. As a matter of fact the wear is very slight.

'The M. A. N. has recently fitted to its large gas engines a valve gear which, besides effecting a more efficient utilisation of the gas at low loads, contributes to the simplicity of the construction as well as to the appearance of the engine.

'The mixture valve and the inlet valve are combined and rigidly connected. They are operated by a system of rolling levers deriving their motion from an eccentric, which at the same time also operates the exhaust valve on the same end of the cylinder. Thus on each end of the cylinder there is now only one eccentric where formerly three were required. The air is regulated at the same time as the gas by an air slide valve rigidly connected to the gas valve. The governor alters the lift of the combined valve by moving a die which is inserted between the upper and lower inlet valve rolling levers, and in this manner the quantity of mixture drawn in is suitably proportioned to the required output. The composition of the mixture is practically the same on all loads, after the valves have once been set for the most efficient combustion, according to the kind of gas used. If the gas changes, then the ratio of air to gas can be readily altered by hand whilst the engine is running, by turning the air slide valve.

'As a highly ignitible mixture is obtained, even at the lowest load, misfires do not occur, and the consumption of gas per BHP hour on light loads is less than in the case of engines fitted with pure mixture regulation.

'The valve discs are now placed on the inside cylinder wall, which is an improvement on the former method, because the valve chambers are no longer subjected to the high explosion pressures and temperatures, and thus the durability of the cylinder is greatly enhanced.

'Ignition is effected in each cylinder end by two ignition plugs, which are actuated by small electro-magnets placed at the points where the ignition has to take place. All points of ignition can be easily adjusted, collectively or singly, whilst the engine is in operation, by a hand-wheel on the switch-gear.

'In the Nuremberg gas engine special oil pumps for the cylinders, stuffing-boxes, and exhaust valves are provided, and it is thus possible to lubricate each point individually according to requirements. The working surface of each cylinder has three feeds, one on the top in the middle, and two at the side. The lateral feeds can be easily cleaned from outside by means of a wire, and the upper oil hole is formed as a non-return valve, which renders its closing up by deposit impossible.

'The forced lubrication of the moving parts is effected from a large oil tank, situated above the engine. From this tank the oil is conducted through suitable regulating valves to the various lubricating points, through pipes of sufficiently large bore to render clogging impossible. The surplus oil is drained off into the basement of the engine-house, where it is collected, automatically filtered, and pumped up into the tank again by means of an oil pump driven from the crank shaft. The rolling levers and eccentrics are, for the sake of cleanliness, lubricated with grease, which is quite efficient owing to the low speed of these parts.

'The pressure required by the cooling water for the cylinder and cylinder cover is about 15 lbs. per square inch (1 atm.). For the piston and piston rods, however, the water requires a pressure of about 45 to 65 lbs. per square inch on account of their reciprocating motion. If, as is generally the case, water of this pressure is available, all points to be cooled are fed direct from a common collecting main. In the event, however, of this not being the case, a pump is provided for the piston cooling water which is worked off the crankshaft, and which produces the required pressure. Each cylinder is fitted with an open water tank, into which the various water drain-pipes discharge freely, this discharge being always in view. The control of the cooling water is well arranged; each discharge is provided with a thermometer and a wheel valve, so that the water temperature of each part to be cooled can be independently regulated to the required degree. To avoid the necessity of altering each of the outlets when the engine is stopped, a stop valve is fitted in the main water-duct, and this valve is opened or closed only during starting or stopping..

'In the case of gas engines from 500 HP upwards, an electric barring gear is provided for bringing the crankshaft into the right position for starting, and for moving the driving gear for examination and cleaning purposes. The engine is then started by means of compressed air at about 150 lbs. per square inch (10 atm.). The compressed air starting valve is first actuated by means of a hand-lever, and subsequently from the lay-shaft by means of a cam. A non-return valve is fitted at the point where the compressed air is admitted into the cylinder. Starting is effected so quickly that, after the second or third revolution, the engine runs without the aid of compressed air. Starting and stopping are controlled also by the main gas-stop valve, the butterfly valves for gas and air on the engine, and the ignition switch.'

Fig. 85 is a longitudinal section through the cylinder of a Nuremberg gas engine of earlier date, and fig. 86 is a transverse section through the valves of this earlier engine to show the modifications made in the later engine.

In the later design the valve pockets are not included in the combustion space; the seated end of the valve cage is carried lower down, so that both inlet and exhaust valves when opened enter the main cylinder.

The projection of the watered cylinder cover, however, acts to prevent a valve falling into the cylinder in the event of its connections giving way. The combustion space also in the recent arrangement is confined to the cylinder, so that the hot gases are contained in one chamber instead of three; the cooling surface is thus reduced, and less heat flows through the casting to the water jacket. By this arrangement heat stresses are reduced to a minimum. It will be noticed also that the valve arrangements are greatly simplified. The two separate mixture controlling valves are dispensed with, and the gas valve is rigidly attached to the charge inlet valve and operated with it.

This dispenses with the two eccentrics and the links and levers required to operate these valves.

The arrangement shown in fig. 86 requires two eccentrics, one to the inlet and one to the exhaust valve. In the improved arrangement one eccentric actuates the links and rolling levers for opening and

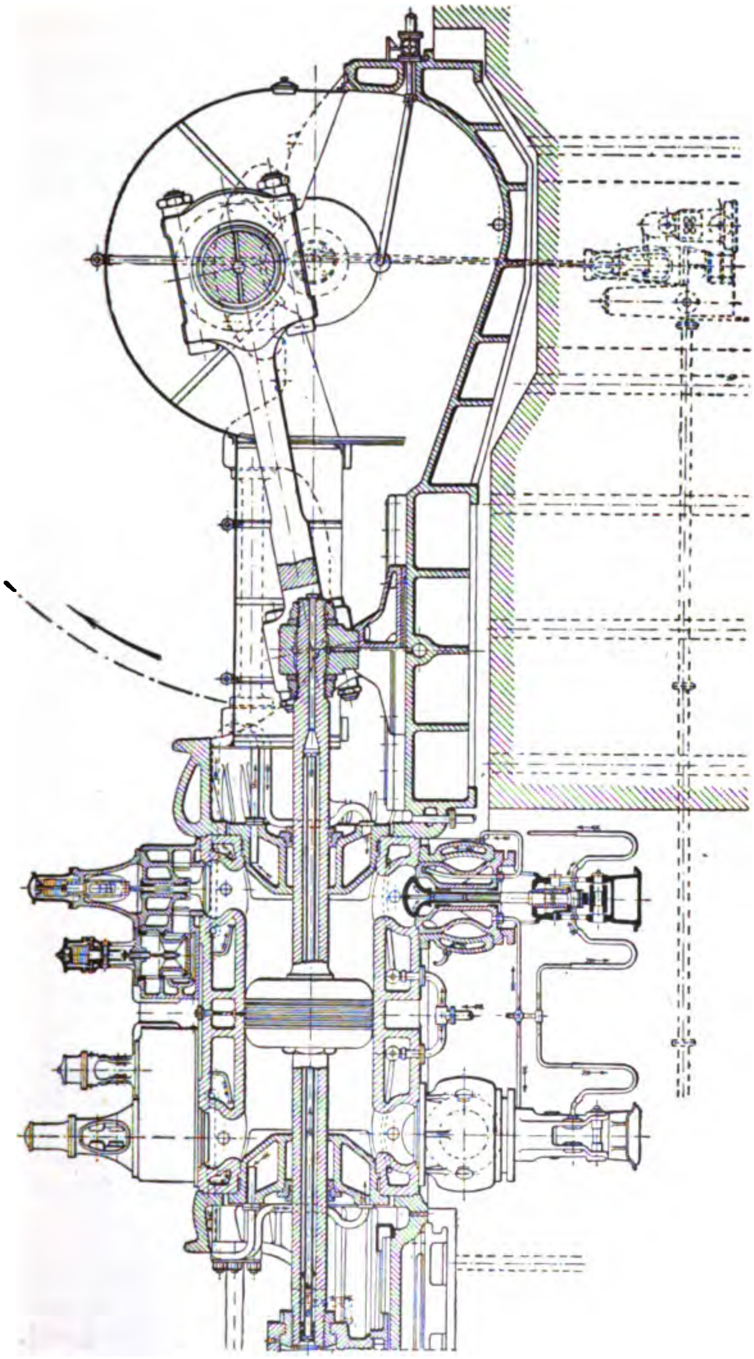

FIG. 85

closing both valves. Altogether the new arrangement requires four valves and two eccentrics per cylinder to do the same work as six valves and six eccentrics in the older design.

The rolling levers used for varying the leverage in opening and closing the valves and also the governing by shifting fulcrum are the same in both.

The whole arrangement of the engine is admirably simple and effective; the only point open to criticism is the electro-magnetic operating gear for breaking contact between the sparking points in the low-tension plug. It seems to the author that this contact gear is very neat and beautiful in action and idea, but somewhat delicate and more easily deranged by tar or dust than the ordinary mechanical break.

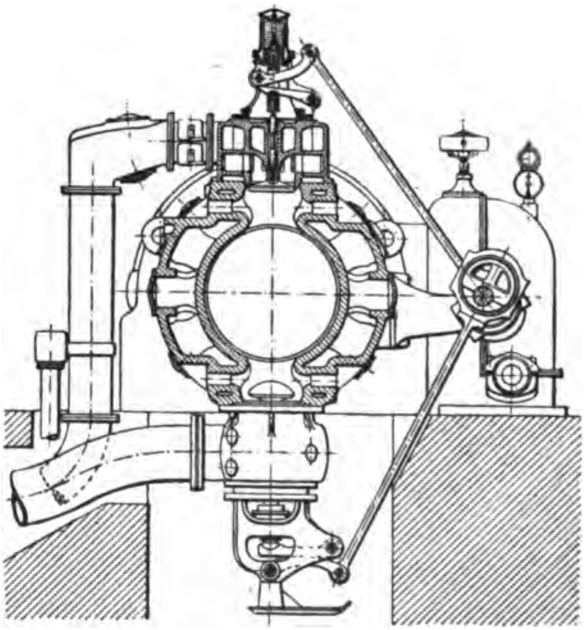

FIG. 86

As to length of time of running continuously night and day without cleaning, the makers give the following information:

'By using well cleaned gas containing not more than 0·009 grains of dust per cub. ft. (0·02 grams per cub. metre) and pure water, requirements which the majority of works can fulfil nowadays, Nuremberg gas engines can run continuously for five months, and longer, without

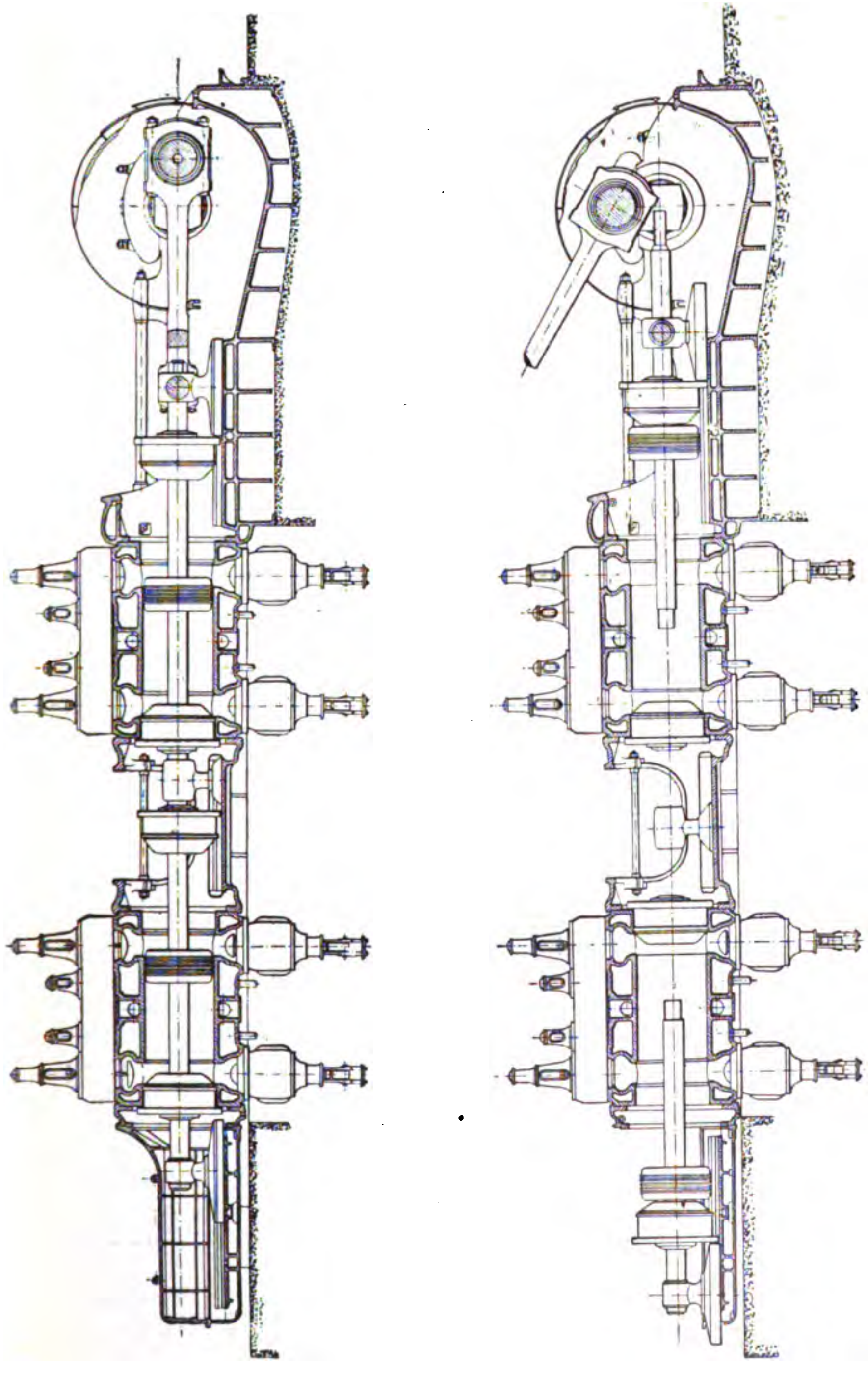

FIG. 87

it being necessary to clean the inlet and gas valves, whilst pistons, cylinders, and cylinder covers require cleaning only after a much longer run.

'A Nuremberg gas blowing engine of 2000 BHP ran day and night in an iron and steel works on blast furnace gas for a period of 19 months. During this time, the gas engine was in operation 98·3 per cent. of the possible working hours, and the stops, 1·7 per cent., were caused by repairs to the blast furnace, and not on account of the gas engine itself. These stops were of sufficiently long duration to enable inspection of the valves, ignition, &c. At the end of the 19 months the engine was in best working order and continued in operation.

'The first double-acting Nuremberg gas engines installed have been working satisfactorily since 1903.'

When cleaning is necessary, however, the arrangement of the cylinder covers and connecting and piston rods permit of ready access both to valves, cylinders, and pistons. The diagrammatic sections given at fig. 87 show clearly how the cylinder covers and other parts are dealt with to enable the interior to be inspected and cleaned.

Professor Riedler has made tests of a 1200 HP Nuremberg gas engine at the Rombach Iron Works, Alsace Lorraine, after the engine had been running without stopping for five weeks under variable loads. The tests were made in September 1904, and particulars are given in Mr. Junge's [1] interesting work on 'Gas Power,' from which the author has prepared the table on p. 133.

The best result obtained by Professor Riedler in these tests was 28·5 per cent. brake thermal efficiency, and as the best mechanical efficiency 83·1 per cent. was somewhat low, the indicated thermal efficiency was high—34·3 per cent.

However, the maximum brake power developed was slightly under the rated power—1186 horse instead of 1200—so that the engine would give still better results with a heavier load. A mean effective pressure of 75·8 lbs. per sq. in. was, however, quite high enough for a cylinder of 33½ ins. diameter.

Mr. Junge states that as the engine was tested immediately after a five weeks' continuous run the results are therefore not the best which it is possible to obtain when working under more favourable (shop test) conditions. But they show a performance that can be absolutely relied upon in actual practice.

The Nuremberg Company give particulars of tests made in February 1908 with two 2400 BHP engines by Professor Langer of the Technical College, Aix-la-Chapelle.

The engines form part of a large installation of 16,600 BHP at pit

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[1] *Gas Power*, by F. E. Junge, M.A., C.E. (Hill Publishing Co., New York).

Anna II near Alsdorf. The engines were supplied with coke oven gas, and they actuated dynamos which were working in parallel.

TESTS OF A NUREMBURG TANDEM DOUBLE-ACTING GAS ENGINE OF 1200 BHP.
(*Riedler*)

With Blast Furnace Gas

Dimensions: *Diameter* 33·46 *ins.* × 43.3 *ins. stroke*

Number of test	I.	III.	IV.	VI.	VIII.	IX.	V.
Revs. per minute	106	105·8	106·3	106·5	106·1	105·8	105·6
Brake horse-power	280	557	871·5	1037	1115	1147	1186
Indicated horse-power	—	807	1146	1312	1359	1388	1427
Mechanical efficiency %	48·5	69	76·2	79	82·1	82·6	83·1
Mean effective pressure in lbs. per sq. in. (average of both cylinders)	—	42·5	60·0	68·7	71·4	73·1	75·3
Consumption of blast furnace gas per IHP hour, cub. ft.	101	100·6	91·5	86·9	87·2	85·5	84·6
Consumption of blast furnace gas per BHP hour, cub. ft.	213	145·6	120·5	109·5	106·3	103·4	101·8
Heating value of gas, B.Th.U. per cub. ft.	88·5	88·5	89·4	89·6	90·7	88·9	87·9
B.Th.U. per BHP hour	18532	12262	10794	9921	9675	9226	8976
B.Th.U. per IHP hour	8984	8452	8214	7829	7937	7619	7460
Brake thermal efficiency %	13·8	20·9	23·7	25·8	26·4	27·7	28·5
Indicated therm. efficiency %	28·5	30·2	31·1	32·6	32·3	33·6	34·3
Duration of test in minutes	33′	28′	29′	26′50″	25′51″	25′51″	25′20″

The engines were guaranteed to consume not more than 7920 B.Th.U. in gas delivered to the supply pipe per indicated horse-power per hour, and the total efficiency from indicated power to dynamo output was also guaranteed not to be less than 79·4 per cent., taking the efficiency of the dynamo at 94·5 per cent.

Professor Langer's results are given by the Nuremberg Co. as follows:

TEST OF TWO NUREMBERG TANDEM DOUBLE-ACTING ENGINES OF 2400 BHP WORKING ON COKE-OVEN GAS. (*Langer*)

—	Test No.	
	I.	II.
Mean test output in kilowatts	1530	1554
Heat consumption in B.Th.U. per IHP hour	6870	6865
Heat consumption in B.Th.U. per KW hour	11,000	10,900
Efficiency attained at full load	0·844	0·857

From these values it appears that in the tests the first engine developed 2162 BHP and the second 2200 BHP. The mechanical efficiency in the first test was 89·3 per cent. and in the second 89·7 per cent.

The indicated thermal efficiency in the first case was 37·3 per cent. and in the second 37·4 per cent. The respective brake thermal efficiencies are therefore 33·4 per cent. and 33·6 per cent.

These are excellent efficiencies; no less than one-third of the whole heat of the gas supplied is given by the engines in brake horse-power.

The Nuremberg Company also use producer gas made from lignite briquettes. and they give the results of tests made by the 'Bayerischen Revisions-Verein' in November 1908, at the paper mills of M. Ellern, Fürth Bavaria, as follows:

Test of Nuremberg Tandem Double-acting Engine of 1200 BHP—Gas from Lignite Briquettes. (*Bayerischen Revisions-Verein*)

Mean output on test	1434·5 IHP
Consumption per IHP hour of briquettes having a lower calorific value of 9000 B.Th.U per lb.	1·12 lbs.
Cooling water: consumption per IHP hour	7·6 gallons
Oil consumption per hour for cylinder lubrication . . .	2·22 lbs.

The total heat given to the producer per IHP hour is thus 10,080 B.Th.U., giving an over-all efficiency from producer to IHP of 25·4 per cent., and, taking the mechanical efficiency of the engine as 85 per cent., a corresponding brake thermal efficiency of 21·6 per cent. From this it appears that the efficiency of the briquette producer is about 75 per cent.

The Nuremberg Co. guarantee their engines to give one BHP hour at full load, on from 8700–9800 B.Th.U. heat in the gas (lower value), but in actual practice they obtain lower consumptions.

The company's experience shows that the engines require 9·3–12·4 grains of oil for cylinder and stuffing-box lubrication, and 6·2–7·5 grains of engine oil for the moving parts, both per BHP hour.

The cooling water is required to absorb about 2800–3600 B.Th.U. per BHP hour, which corresponds to 7·7 gals. of cooling water having an inlet temperature of 59° F. (15° C.).

By installing cooling plants the consumption of water may be reduced to 0·44 gals. per BHP hour.

For purifying the gas about 2·2 gals. per BHP additional is required.

Ehrhardt & Sehmer Engines

Messrs. Ehrhardt & Sehmer of Schleifmühle are successful builders of large gas engines of the tandem, double-acting, four-cycle type.

In their early engines the cylinder and water jacket casing were cast in one piece, but their later engines have built-up cylinders.

Messrs. Galloways of Manchester now build these engines in England. Fig. 88 shows a general view of a recent double-acting tandem Ehrhardt & Sehmer gas engine of about 2400 HP.

Fig. 89 is a longitudinal section.

Fig. 90 is a transverse section through the valves, showing also the valve operating gear.

Fig. 91 is a longitudinal section through the cylinder on a larger scale.

The diameter of the cylinder is 51·2 ins. and the stroke 52 ins.

The speed of rotation is 94 revolutions per minute.

It will be observed that the cylinder is formed of two main castings bolted together transversely by flanges at the middle. The two parts are similar in all respects, and each half carries the necessary valve pockets, ports; and facings. The portions of the water jacket near the valves are formed by casting the ends and water casings together with the cylinder, and the intermediate water space is formed by means of a light casing bolted longitudinally to go on in halves. The water joints are formed by rubber. To produce a continuous surface within the cylinder an inner liner is used which is held by a projecting flange at the middle, which flange is clipped and firmly held between the two central flanges in a groove formed in the two ends as shown. The liner is thus held in the middle, and it is free to expand at its outer ends. It serves the purpose of providing a continuous cylindrical surface for the piston, shielding the main casting from the heat of the gases and providing an easily renewable liner which can be replaced at small cost, when the cylinder requires re-boring. The use of this liner seems to the author to offer considerable advantage in large cylinders with thick walls. The thin inner liner takes a higher temperature without danger from heat stresses, and the thick main cylinder gives it the support necessary against the bursting stresses due to the explosion pressures. To some extent the combination resembles in its action a wire-wound gun, and it seems likely to greatly diminish the risk of fracture by heat and bursting stresses due to explosion pressures.

The cylinder without its covers is thus formed of five separate pieces: the transversely divided cylinder two pieces; the inner liner one piece; and the loose water jacket casing and the central part of the jacket two pieces.

To some extent this built-up cylinder resembles the earlier form of 2000 HP Deutz engine cylinder shown in section at fig. 77. In the Deutz cylinder the centre working cylinder for the piston is cast in one piece, and the cylinder ends carrying the valve pockets and

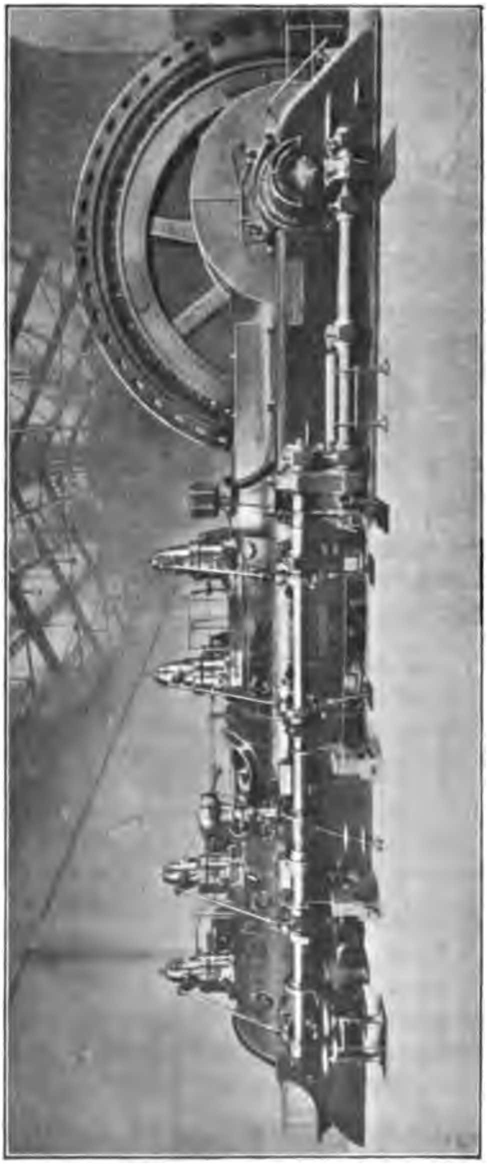

FIG. 88

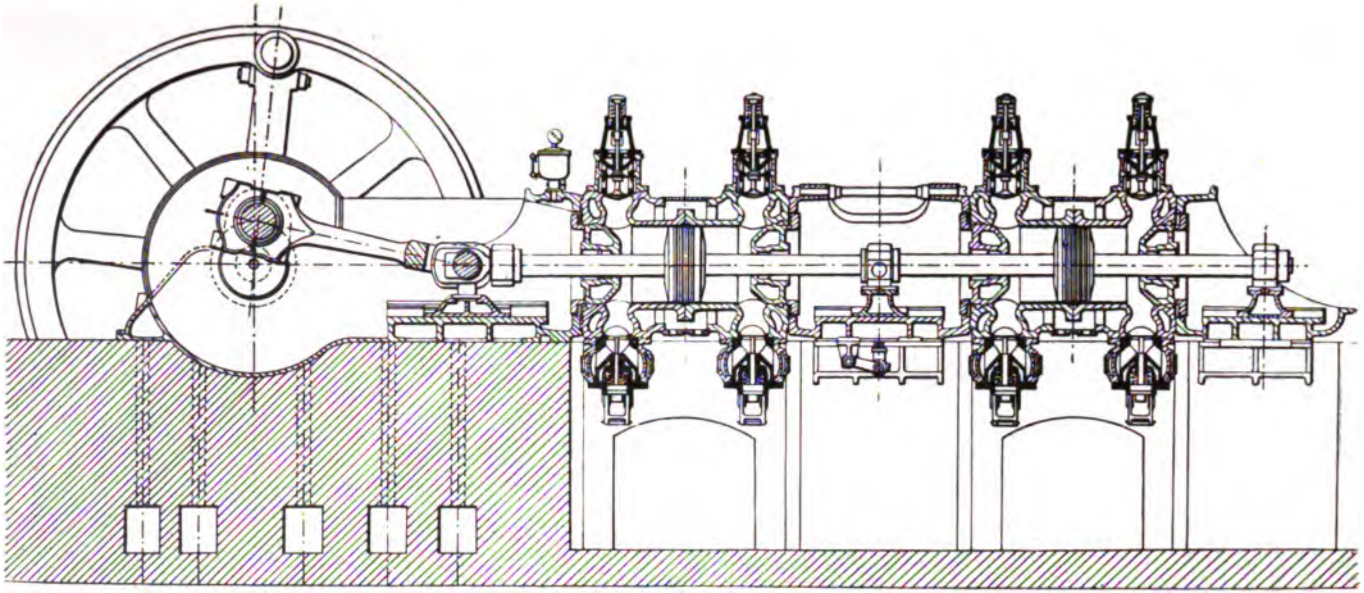

FIG. 89

ports are cast separately and bolted on by transverse flanges and bolts. The centre part of the cylinder is here also jacketed by means of a clamped-on casing. In both Ehrhardt & Sehmer and Deutz cylinders the expansion of the inner cylinder is free of constraint from the water-jacket casing.

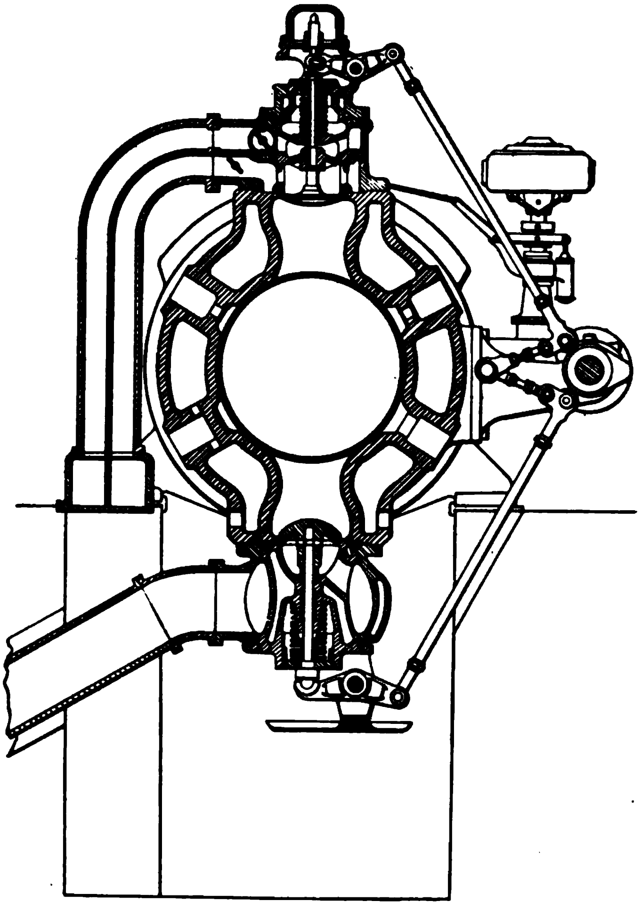

Fig. 90

The Ehrhardt valves, inlet and exhaust, open into pockets leading to the cylinder, and they are both actuated by one cam; the gas valve is carried on the inlet valve stem and it is spring seated, so as to compensate for any wear on the main valve seat, and also to allow of a small amount of opening on the inlet valve to admit air to the cylinder before gas mixture passes into it. The governing is accomplished by means of a plain butterfly throttle, as seen at fig. 90.

An air throttle is fitted which enables the adjustment of gas and air to be made by hand, and also permits of governor control to keep up the strength of the mixture at light loads.

Messrs. Ehrhardt & Sehmer do not water-cool the exhaust valve; they find that if the engines be run with sufficiently dilute mixtures, and sufficiently great ratios of expansion, the exhaust temperatures are so low that no trouble is experienced even in the very largest engines.

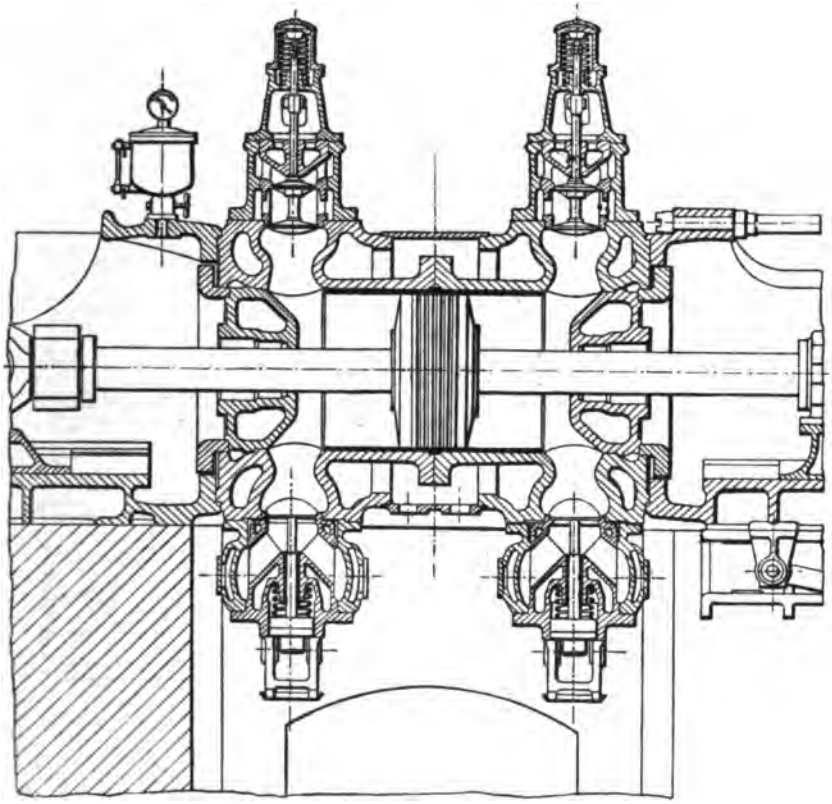

FIG. 91

The valve sleeves carrying the exhaust valves are pushed up within water-jacketed casings, and the sleeves are themselves fully water jacketed. The valve sleeves can thus be removed without disconnecting the exhaust pipes. The air and gas inlet valve sleeves are also carried in casings bolted to facings on the cylinder, and are also easily removable. The air and gas are led through a divided pipe to the air and gas spaces, and they do not mix until they reach the inlet valve.

Fig. 92 shows in longitudinal section an earlier engine cylinder by this firm, in which internal cylinder and water-jacket casing, together with all valve pockets, are cast in one piece. Here, too, separately operated and controlled gas and air valves supply the main inlet valves and increase complication.

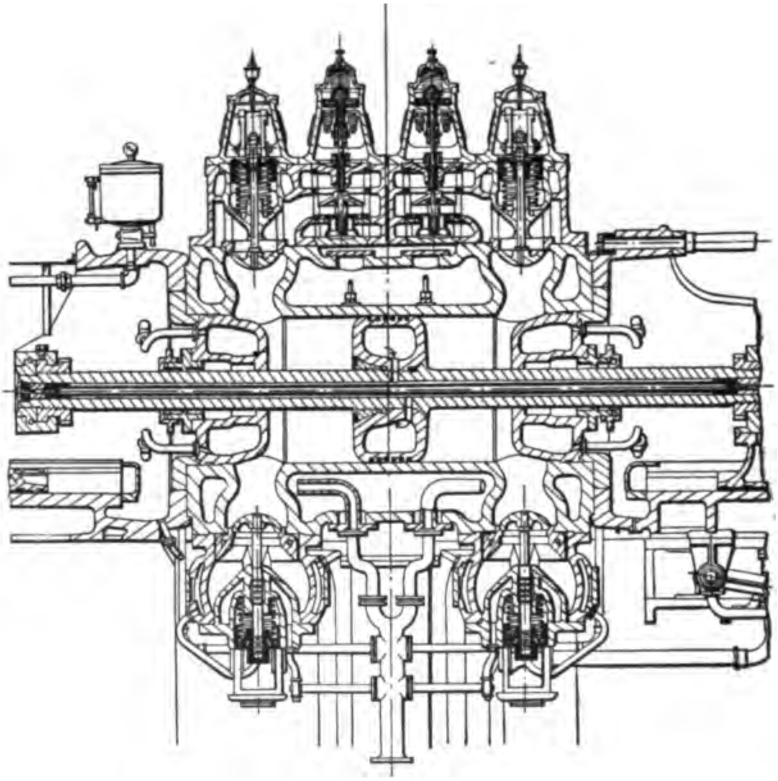

FIG. 92

This cylinder is 31 ins. diameter by 35·5 ins. stroke.

In his valuable book on gas engines[1] Mr. Mathot gives the following account of a test of this engine :

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[1] *The Construction and Working of Internal Combustion Engines*, R. E Mathot. Constable & Co., London.

'In 1906 a test was mutually arranged between the engineers of Ehrhardt & Sehmer and those of the Konigliche-Bergenspection at Heinitz Saarbruck, upon a four cylinder, double acting 600 HP engine.

'After four months of constant work, without any previous cleaning, this engine was put under test with coke oven gas of about 450 to 470 B.Th.U. lower heating value per cub. ft. It showed a consumption of 8000 B.Th.U. per BHP hour. The mechanical efficiency under the load carried was 83 per cent. The engine was new, and was tested with a three-phase dynamo mounted on the engine shaft.

'The principal details were as follows:

Diameter of cylinder	24·5 ins.
Stroke of piston	29·5 „
Diameter of piston rods	6·8 „
BHP average	520
Revolutions per minute	150

'It will be observed that the thermal efficiency per BHP was nearly 31 per cent., or, if based on IHP, 37·5 per cent'

In an interesting paper, 'Large Gas Engines and their Troubles,[1] Mr. Frank Foster discusses the stages of the development of the Ehrhardt & Sehmer cylinder as follows:

'Fig. 93 shows a typical large gas engine cylinder as formerly made by Messrs. Ehrhardt & Sehmer of Saarbrucken, Germany. This cylinder, like a great many other Continental designs, was cast in one piece. There were two concentric barrels tied together by the two

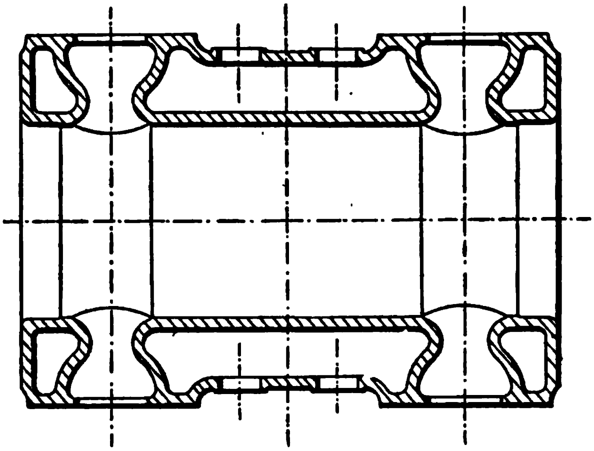

Fig. 93

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[1] Paper read at the Manchester Engineering Exhibition, 1910.

end flanges and the neck pieces of the four main valves, the four igniters, and the four air starting valves, not to mention any ribs there might be connecting the two barrels.

'Such a cylinder has two primary defects. In the first place, the parts are too rigidly tied together to permit of the casting contracting freely in the mould when cooling. Hence such cylinders are never obtained without initial casting stresses of unknown amount. In the second place, when the cylinder is working, the outer barrel is normally cooled by the water jacket and the air, so that it practically does not expand. The inner or working barrel, on the other hand, becomes hot by contact with the gases and expands. The expansion of the inner barrel is resisted by the cold outer barrel. These two barrels pulling in opposition put the outer one into tension and the inner into compression, and tend to break off the neck pieces at the junctions with the barrels. In practice cracks are commonly started at these junctions, and after a time, in spite of ductile straps of metal riveted across the crack, the cylinder has to be removed. In some designs this weakness is increased by omitting to bellmouth the necks at the junctions, such omission not only causing mechanical weakness, but also, by reason of the scour against the sharp corner of metal at the junction, leading to excessive heat stresses.

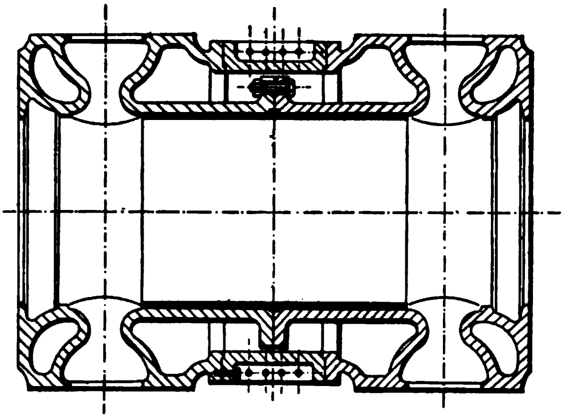


FIG. 94

'The existence of these casting and heat stresses is fully recognised by makers, and much experimenting and scientific care has been expended in trying to overcome these difficulties. Apart from the choice of special foundry mixtures for the castings and the adoption of great care in the details of design and moulding, there are two ways of attacking the problem.

'One, which appeals particularly to Continental designers, is that of balancing the casting stresses against the expansion stresses. Stated thus simply, this method seems highly scientific and satisfactory. Looked at more closely, it bristles with difficulties, although it must be confessed that Continental makers have carried it to a considerable degree of perfection. The difficulties in the way of a satisfactory result are serious, and, however skilful the designer may be, there must always be some uncertainty. Thus, initial casting stresses are uncertain in amount. By pouring the metal at one temperature and by care and uniformity in the moulding—particularly as regards the cores—so that the mould always offers the same total and local resistances to the contraction of the metal, a very great step in the direction of uniformity is made, but with the best of designs and the most rigid adherence to scientific foundry methods it still remains impossible to estimate the value of the initial casting stresses, or to maintain them at one figure in successive castings. Further, it is not possible to estimate accurately the expansion stresses. Experiments have been conducted with great care, but the results are little more than general indications of what is taking place; and in any case these stresses vary with the working conditions in the cylinder, and with the effectiveness of the jackets, both as a whole and in their sections. Hence, by this method one has to balance one unknown and variable stress by another unknown and variable stress.

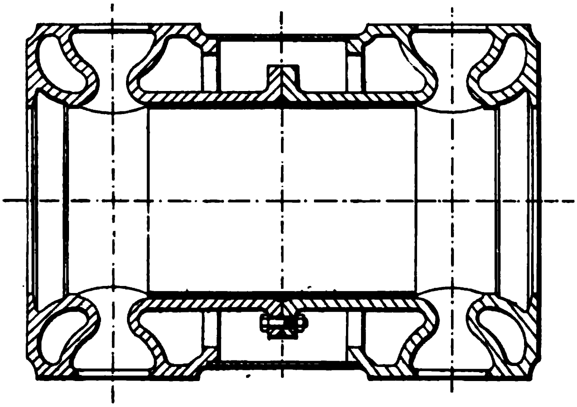


FIG. 95

'The other method of attacking the problem will appeal strongly to English engineers. Instead of attempting an unsteady balance between unknowns, the unknowns are eliminated as far as possible. Thus in the Ehrhardt & Sehmer engine, as made by Messrs. Galloways of Manchester, the casting stresses are reduced to a minimum by adopting a built-up construction, as illustrated in fig. 95. The cylinder body is split transversely and is cast in symmetrical halves. Each approximates to a double cylinder with a cross-section of flat U-shape. Obviously the U section permits very free contraction in the mould, and casting stresses are reduced to a minimum. Fig. 96 illustrates one-half of an Ehrhardt & Sehmer cylinder, showing the casting position in the mould. In its completed form the outer barrel or jacket casing is a rolled steel band in halves clipped into the cylinder body through rubber joint rings. These rings form an expansion joint, so that the inner barrel is free to expand, and in practice a movement of as much as 2 mm. takes place. It is important that the jacket should not be bolted rigidly to the cylinder body or ends. In one of their designs intermediate between the old single-piece cylinder and the modern built-up type Messrs. Ehrhardt & Sehmer did adopt a cast-iron jacket bolted to both cylinder ends, as shown in fig. 94, but it did not permit the cylinder body to expand, and was abandoned. Strangely enough this defective method is still in use by some makers. A somewhat similar construction to that adopted by Messrs. Ehrhardt & Sehmer is being built by some of the American firms, and evidence of the success of this method of attacking the problem is to be found in the number of makers who are trying built-up designs. One of the incidental advantages of a cylinder of the type illustrated in fig. 95 is that should one cylinder end crack it is not necessary to replace the whole cylinder, but only the defective half.

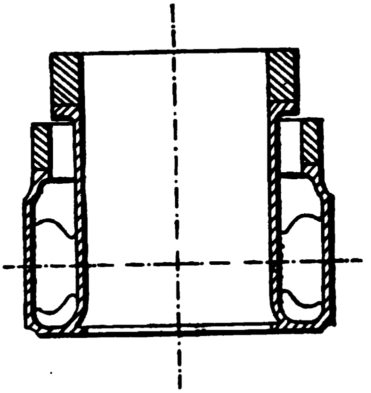


Fig. 96

'Before leaving this part of our subject, it may be as well to refer to the liner shown in Fig. 95. Objection has been taken to liners for gas-engine cylinders on the ground that they work loose. In this case the liner is held between the two cylinder ends by a collar, and cannot work loose. This liner has two functions. It is intended to act as a wearing liner, which can be bored out when necessary, and finally renewed without replacing the cylinder body. Also it acts as a heat shield to the cylinder barrel.'

RICHARDSON, WESTGARTH & CO.'S COCKERILL ENGINES

Messrs. Cockerill and Messrs. Richardson, Westgarth & Co. have also appreciated the advantages of the double-acting tandem gas engine, and they now build all their gas engines of this type, as it has proved more reliable and less expensive to build than the original single-acting open trunk engine.

Fig. 97 is a photograph showing one of their early engines of this type in the foreground and two of their earlier tandem single-acting engines in background. In the double-acting engine at first they followed the valve arrangement of the older engines described at figs. 70 *a*, *b*, *c*. In this design it will be remembered that both inlet and exhaust valves are placed below the cylinders, and while a neat, smooth appearance is thus given to the upper part of the cylinders, yet the construction involves the very long valve chamber with its want of symmetry already discussed. In later engines this arrangement was changed, and the ordinary type followed, in which the air and gas inlet valves are placed above the cylinder and the exhaust valves below.

Fig. 98 is a photograph of such an engine.

Figs. 99 and 100 are respectively side elevation part in section and plan part in section of an engine of this type. Fig. 101 is a transverse section on a large scale through the valves.

Fig. 102 is a longitudinal sectional plan of the cylinder and its connections on a larger scale.

The engine bed consists of two long cast-iron box girders bolted to a front bed casting which carries the main bearing and the cross-head slides.

The cylinders are cast in one piece, inner or main cylinder and outer or water jacket casing, with all the necessary ports and valve pockets and facings.

To make certain that casting stresses neither put the internal nor external cylinders in compression or tension, the outer cylinder or casing is parted in the lathe at one point, so that the inner cylinder is freed from the outer; this cut is made towards the rear end of each cylinder, and a flexible water joint is introduced. The cylinder is bolted, by means of suitable facings and studs, by its jacket casing to the side girders, as shown in fig. 102; the front end is held longitudinally between girder lugs and fastened by wedges as shown; the rear end is similarly fastened to the girders by studs, but the facings are free longitudinally to slide over one another by allowing some play in the stud holes passing through the girder. All the power and explosion stresses are thus taken by the projecting facings and lugs at the front ends. On the back stroke the pressure of the explosion

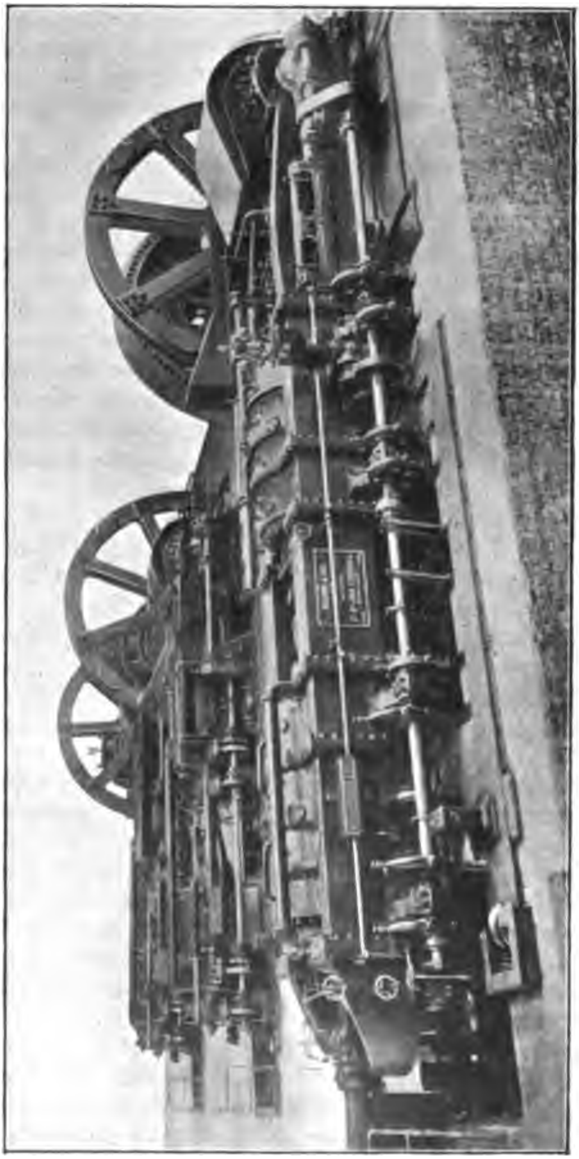


Fig. 97

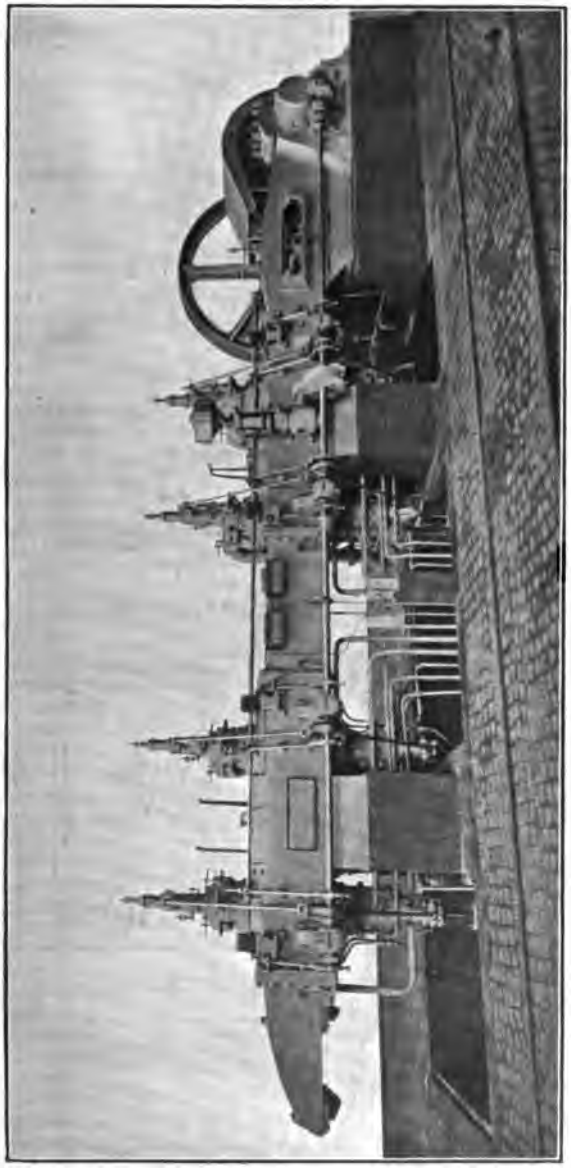


Fig. 98

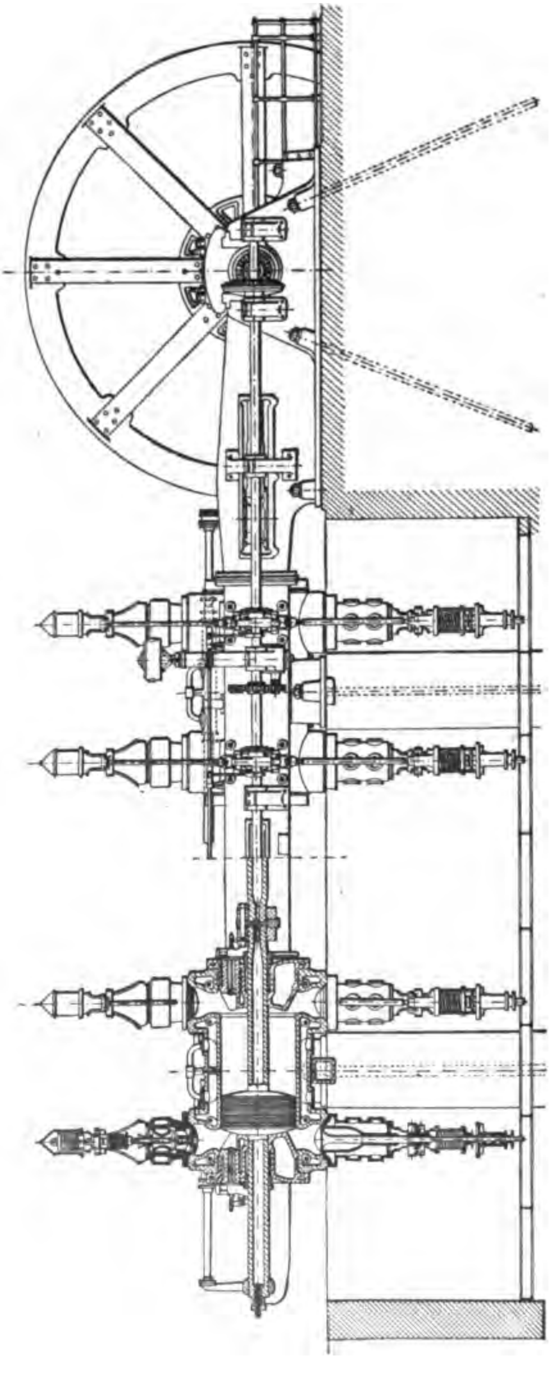


Fig. 99

acting on the cylinder cover is taken by a short path through the metal of the casing to the girder lugs, and this is done without putting the inner or principal cylinder into longitudinal tension by means of the long bolts which hold on the cylinder covers and compress the inner cylinder longitudinally. Longitudinal tension stresses are thus prevented in great measure, if not entirely, from affecting the inner or working cylinder.

This ingenious method of constructing the cylinder and connecting it with the bed girders operates very successfully.

Referring to figs. 99 and 100, it will be seen that the charge inlet valves are arranged above the cylinders, and they open into pockets leading into the cylinder; the exhaust valves are placed below, and open into similar pockets. The water is introduced into the piston rod at the front end at the crosshead by a system of sliding tubes; it passes through the piston rods and both pistons, and is discharged at the rear crosshead into a trough.

In this particular engine the compressed air starting valves are carried by the cylinder covers as shown in the sectional part of fig. 99. `In other modifications they are arranged in the cylinder proper as shown in the cross-section, fig. 101.

Fig. 101 clearly shows the valve arrangements. The main charge inlet valve carries on its spindle a cylindrical valve with ports, which acts as a guide for the spindle and also controls the air supply ports.

The gas inlet valve is connected with the main inlet valve and is carried on a hollow spindle through which passes the main valve spindle.

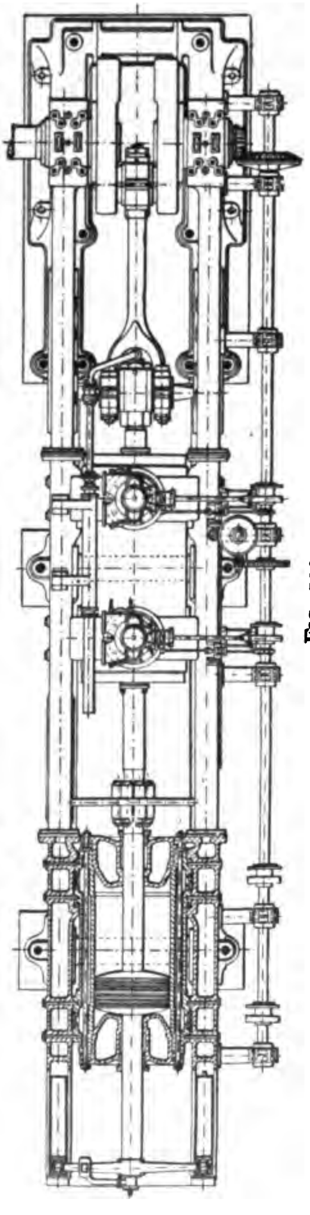


FIG. 100

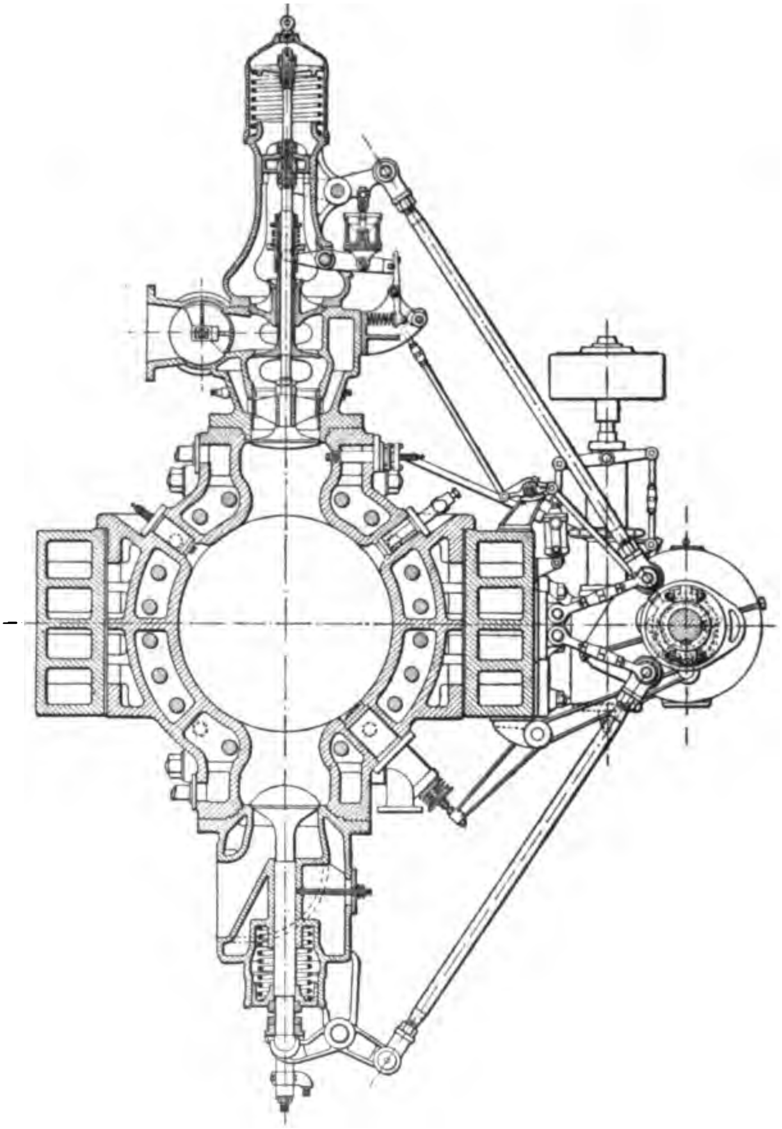


FIG. 101

The gas and air valves thus operate independently of each other. The main inlet valve is opened by cam, link, and lever, in the usual way, and at the beginning of the stroke the air passages only are opened to the cylinder; air is thus admitted before gas. Meantime the upper operating lever in lifting at its outer end pulls up a piston fitted within a cylinder pivoted to a gas valve lever, which lever connects to the gas valve sleeve by side links and a spring. When the piston is pulled upwards the pressure within the small cylinder is reduced, and the external atmospheric pressure tends to move the gas valve lever so as to lift the gas valve off its seat; the lifting, however, is prevented by a trip lever, which is controlled by the governor. So long as this lever engages the gas valve lever by its catch plate the gas valve remains closed. At a point determined by the governor this trip lever liberates the gas valve lever, and the vacuum cylinder at once operates and opens the gas valve very quickly. Gas then flows into the cylinder with the air during the remainder of the suction stroke, and at the end of that stroke the gas valve closes slightly before the main inlet, operated by the main charge inlet lever acting through the vacuum cylinder and piston with its connected gas valve lever.

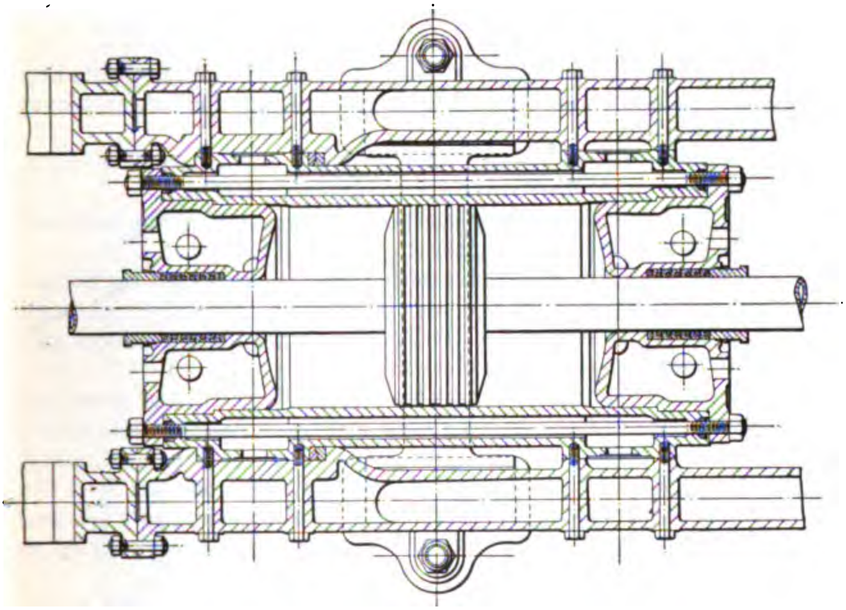


FIG. 102

In this way a slight flow of air displaces any combustible mixture remaining above the inlet valve, and so provides that pure air shall enter the cylinder at the beginning of the next charge.

By this device the governor admits gas later and later, and secures that the large valve pocket shall always be filled with an ignitible mixture of gas and air to produce an impulse at the lightest load. It will be seen that two electrical igniters are used, one placed close up to the inlet valve and the other in the body of the cylinder, intermediate between the charging port and the middle of the cylinder. The top lever is liberated by a rocking cam surface operated from an eccentric on the valve shaft. The position of the trip is determined by the governor movement.

The exhaust valve opens into a lower pocket, and it is operated by the same cam which actuates the inlet, by means of a link and lever clearly shown.

The compressed air starting valve is seen in the section to the right of the exhaust valve, and it is operated by lever and cam.

In this engine the quality method of governing is adopted; the compression remains fairly constant.

The first engine of the double-acting tandem type used by Messrs. Richardson, Westgarth & Co. was erected at their Middlesbrough works in 1903. It was of 500 BHP, driven with Mond producer gas, and was direct coupled to an electric generator.

The cylinder diameter was 23·5 ins., stroke 35·5 ins., and speed 130 revolutions per minute. In 1904 they had two tandem double-acting engines in operation of 800 BHP each, one at Messrs. Cochrane's, Middlesbrough, and the other at the Frodingham Iron and Steel Co. These engines have tandem double-acting cylinders each 29·5 ins. and 35·3 ins. stroke, speed 120 revs. per minute; they are direct coupled to generators.

Fig. 103 shows indicator diagrams taken from the 800 HP engine at Messrs. Cochrane's and handed to the author by Mr. Westgarth in 1904. The particulars are marked above the diagrams.

The compression pressure on these diagrams is about 150 lbs. per sq. in. above atmosphere; maximum pressure of explosion on any of the cards 360 lbs. above atmosphere; mean effective pressure average of the four cards 72·6 lbs. per sq. in.; and maximum combustion temperature about 1400° C.

The IHP at 98 revolutions per minute was 810. Messrs. Richardson, Westgarth & Co. have also built engines of this type of 1200 BHP; revolutions 120 per minute, cylinder diameter 31·5 ins., and stroke 35·5 ins.

The line of development of the large four-cycle engine on the Continent has been sufficiently indicated in the descriptions which

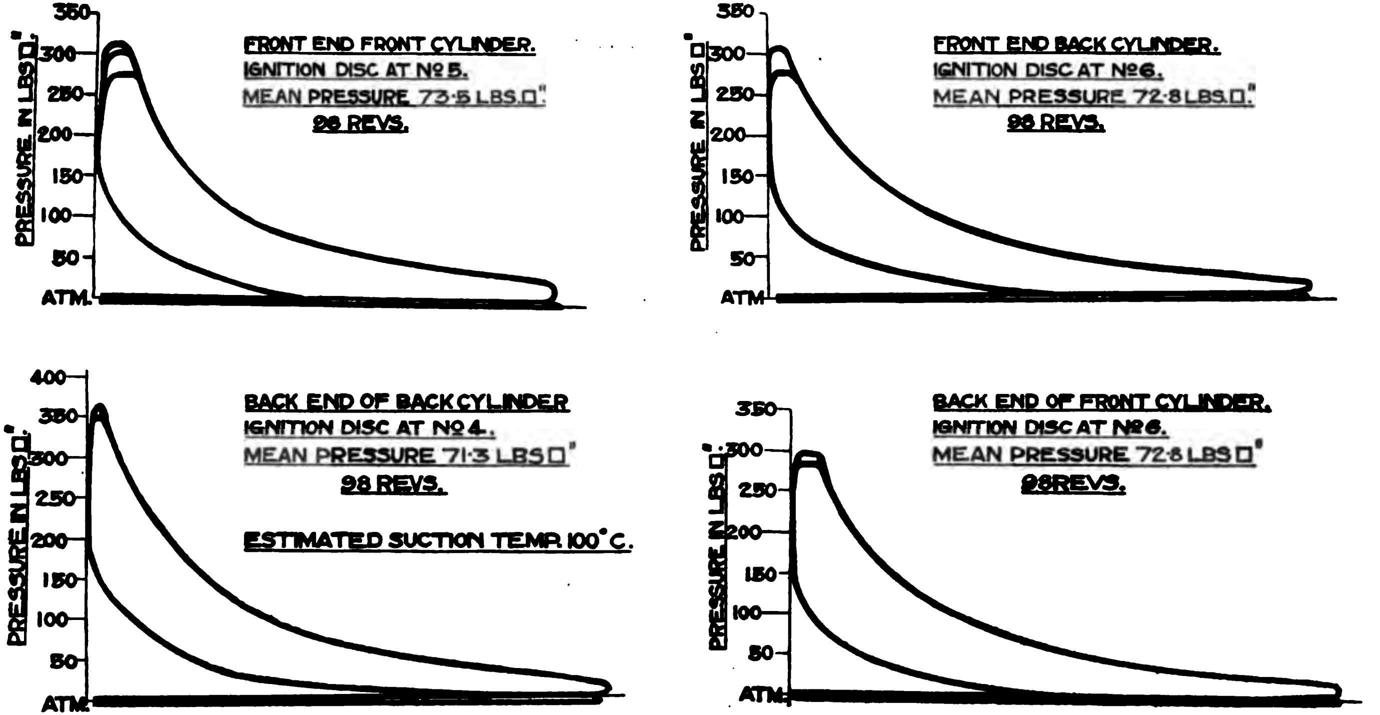

350
300
250
200
150
100
50
ATM.
PRESSURE IN LBS□"
FRONT END FRONT CYLINDER.
IGNITION DISC AT No 5.
MEAN PRESSURE 73·5 LBS.□"
98 REVS.
FRONT END BACK CYLINDER.
IGNITION DISC AT No 6.
MEAN PRESSURE 72·8 LBS.□"
98 REVS.
400
BACK END OF BACK CYLINDER
IGNITION DISC AT No 4.
MEAN PRESSURE 71·3 LBS□"
98 REVS.
ESTIMATED SUCTION TEMP. 100° C.
BACK END OF FRONT CYLINDER.
IGNITION DISC AT No 6.
MEAN PRESSURE 72·8 LBS□"
98 REVS.

Fig. 103

have been given of the Cockerill, Deutz, Nuremberg, Ehrhardt & Sehmer, and Richardson-Westgarth Cockerill designs, but many other able engineers successfully build such engines. Messrs. Haniel & Lueg build cylinders as large as 51 ins. diameter; at one time a twin tandem engine giving 4000 BHP by them was considered to be the largest on the Continent. Later Messrs. Thyson & Co. installed at their Mulheim works a tandem double-acting engine of which the two cylinders were each 48 ins. diameter, and stroke 55·5 ins., which at 94 revolutions per minute gave 2600 BHP. A twin tandem of this type would thus have given 5200 BHP.

AMERICAN LARGE GAS ENGINES

Relatively few large gas engines have been as yet installed in England, so that in this branch of industry the Continental engineers occupy a decisively leading position, and American engineers provide a good second.

The Nuremberg type is followed in America by the Allis-Chalmers engines, which are built by them up to 3000 HP. A large installation by them consists of seventeen twin tandem double-acting engines, each unit giving 2500 kilowatts, at the works of the Indiana Steel Company, Gary, Ind. The complete set is thus capable of developing 51,000 HP. The Nuremberg type is only departed from by the adoption of overhung cranks and by small variations in ignition and valve gear.

The Snow Steam Pump Company took up the question of the large gas engine in 1904 and they have departed in many particulars from standard Continental designs. They have built many large engines, which in twin tandem double-acting units give as much as 5400 BHP per unit. In one power-house of the San Mater Power Company they have in operation four engines giving a total of 21,600 BHP. Each unit contains four cylinders, each of 42 ins. diameter by 60 ins. stroke, which develop 5400 BHP at 90 revolutions per minute. Each cylinder thus develops 1350 HP.

In a smaller twin tandem engine the four cylinders are each 39 ins. diameter and 54 ins. stroke, and at 90 revolutions per minute they develop 750 horse per cylinder, or 3000 HP total.

The cylinders are of the built-up type cast in two end pieces and connected together in the middle by flanges and bolts. The water jacket casing is cast with the cylinders at the ends but stopped off in the centre, the water jacket casing being completed by a clamped on portion which is made watertight by jointing which allows of free expansion. The valves are carried in chambers cast from the sides of the cylinders. Fig. 104 shows a longitudinal section through the valves and valve chamber, together with a transverse section through the cylinder and water jacket having the valve arrangement in elevation. The main inlet valve casing is arranged above the side chamber which leads by a port into the cylinder, and it is actuated from the valve shaft by the short lever shown. The exhaust valve is placed underneath, and opens into the same chamber; it is actuated by a similar short lever. The gas and air are separately controlled by a double seated valve, which is raised from both seats by a rocking lever connected to the top of the main inlet valve stem. This rocking lever lifts the spindle of the double seated valve by a link and hook (see cross section), which engages a catch plate connected to the stem. The double seat valve is raised when the main valve opens, and air is first admitted, and then gas flows into it. This is effected by causing the gas valve to rise at first through a restricted conical portion before it opens fully. A rotating shaft controlled by the governor pulls out the hook link at a position determined by the load and the double seat valve then drops to close gas and air. The quantity of mixture is

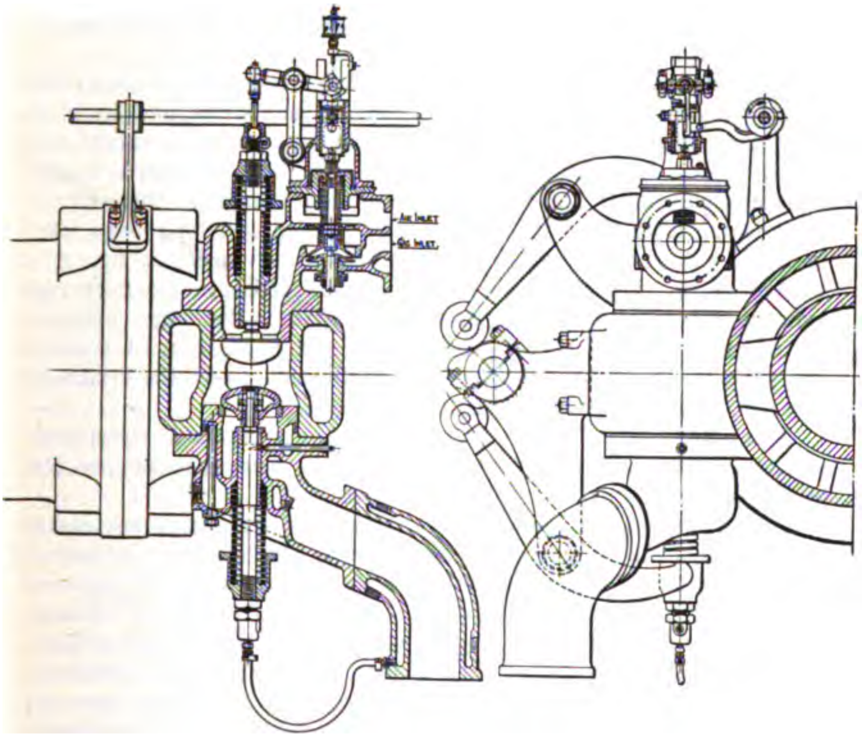


FIG. 104

thus determined by the governor, and the proportion of gas to air is kept constant. The conical valve under the gas valve is used to adjust the gas supply.

The governor-controlled shaft rotates at the same speed as the engine shaft and is driven through a train of bevel gears of the differential type; the governor controls this differential gear, and so determines the point at which the cam trips the hook lever, and so causes the closing of the gas and air supply.

The arrangement of the watered exhaust valve is clearly seen in the section.

The side arrangement of exhaust and inlet valves has considerable advantages in that it permits a continuous foundation and bed. It also allows very direct operation from the cam shaft by simple bell crank levers, and dispenses with the long links required in the Nuremberg design. The valves are also more accessible than in the more usual method. The side chamber design has long been used in small gas engines for exhaust valves, as is shown at fig. 7, a section of an early Otto engine; it is also common for inlet and exhaust valves in many vertical gas engines. For large cylinder engines the importance of symmetry in the disposition of pockets and ports has caused the majority of engineers to prefer the vertically opposed type of inlet and exhaust valve pockets.

The Snow Pump Co., however, have great experience in these large engines, as they had turned out of their works more than 140,000 HP of engines of over 500 HP up to the end of 1911.

The American Westinghouse Company also build large horizontal gas engines of the tandem double-acting type which follow Continental practice in the design of cylinders and bed and in the vertically opposed arrangement of inlet and exhaust valves. The inlet and exhaust valves on top and bottom of the cylinder are operated by a single eccentric, with links and rolling levers. The inlet valve, which is of the ordinary conical seat type, carries on its spindle a balanced cylindrical valve which opens and controls both gas and air inlets. This valve can be rotated round the spindle by the action of the governor, and it has inclined gas and air ports corresponding to similar ports in the valve box. When the governor holds the valve in the full load position the ports are full open, and gas and air are admitted to give a full charge. As the governor rotates the cylindrical valve both gas and air are cut off and the charge weight reduced, while the proportion of gas and air is kept constant. The sleeves are operated by an oil relay device, so that the load on the governor is reduced. The cylinders are of the built-up type, and an expansion joint is provided to free the water jacket casing, the arrangement being somewhat similar to that of the Deutz double-acting engine.

Among other makers of large four-cycle engines may be mentioned the William Tod Co., of Youngsten, Ohio; four twin tandem double-acting engines have been installed by them at the Ohio works of the Carnegie Steel Company to actuate blowers. The power cylinders are 42 ins. diameter by 60 ins. stroke, and the rated speed at full load is 75 revolutions per minute, when each engine consisting of four cylinders develops 3000 HP.

The blowing cylinders are 80 ins. diameter.

The power cylinders are of built-up type, cast in two pieces and joined together in the middle of the cylinder by flanges and bolts in the manner of the Ehrhardt & Sehmer and other Continental engines, the jacket casing, too, is free from the cylinder, and has an expansion joint like the Deutz engines. The explosion and working stresses are taken by long steel bolts running from end to end of the engine frame.

The inlet and exhaust valves above and below the cylinder are actuated from one eccentric by links and the now common rolling levers. Governing is performed by the quality method, in which nearly constant compression is maintained. The governor valve is of a peculiar disc type.

Very general attention is now being paid to large-cylinder gas engines by American engineers, but the preceding short sketch illustrates sufficiently the general trend of American design.

The American Westinghouse Company have paid considerable attention to inverted vertical gas engines, and have installed many sets both in America and in England; their practice, however, does not greatly differ from English design.

SINGLE-ACTING TANDEM VERTICAL FOUR-CYCLE ENGINES

The English Westinghouse Company have naturally been influenced by British ideas as to gas engine design, and among other prepossessions or perhaps prejudices they have felt all the English engineers' repugnance to large double-acting cylinders with watered pistons and piston rods. Accordingly they have attempted to produce large power engines in which the cylinder dimensions were kept sufficiently moderate to dispense with watering, the power being obtained by relatively high speeds of rotation and multiplication of the number of cylinders employed.

This method of solving the high-power problem had been attempted before by Messrs. Burt & Co. of Glasgow about 1894. A description of their engine is given in the sixth edition of this work, published in 1896, p. 340, in which the author gives a vertical section and indicator diagrams. The present author, writing in 1895, there describes the engine as follows:

'Messrs. Burt & Co. have recently built a high-speed gas engine, of which a vertical section is given at fig. 105; it is especially interesting as it approaches so closely to steam engine lines. Two pistons, 1 and 2, are arranged tandem fashion on the same piston rod, 3; a common connecting-rod, 4, serves for both and actuates a crank. The upper sides of the pistons are used for the power impulses, the lower sides moving air idly to and fro; both pistons operate on the Otto cycle, but the impulses are arranged to alternate. The crank thus gets an impulse at every revolution when the engine is under full load. The crank shaft carries a wheel, 6, gearing into a wheel, 7, from which the piston valve is driven at half the number of strokes of the main crank. The action and function of these piston valves

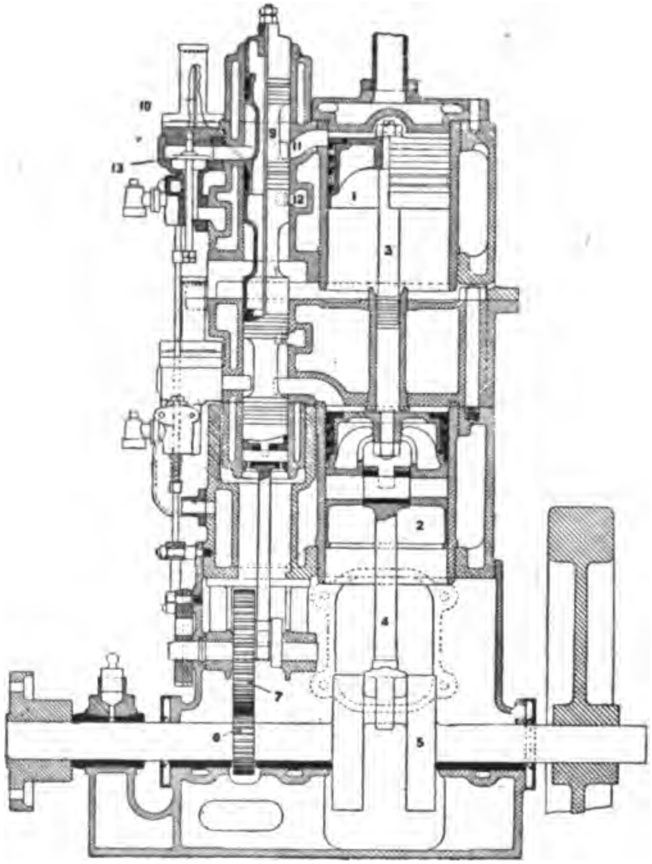
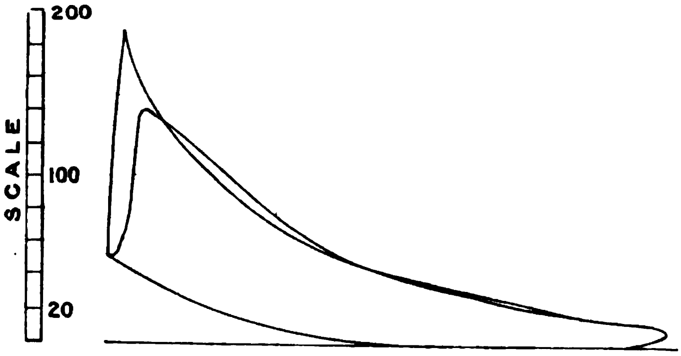
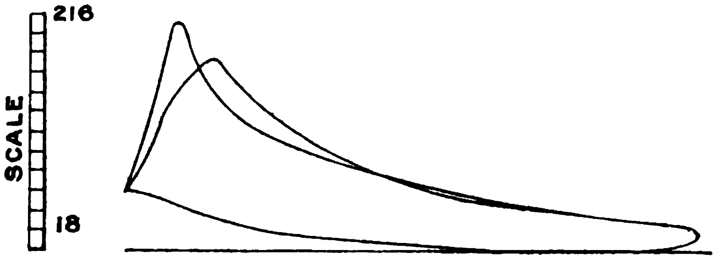


FIG. 105.—Burt's High Speed Otto Engine (vertical section)

are very peculiar. The pistons 1 and 2, it will be seen, approach their cylinder covers as nearly as steam engine pistons, and the main combustion space is formed by the ports and passages leading to the valves, and also by the annular space formed between the piston valve stems 9, and the cylinder. When the upper piston 1 is in the position shown in the figure, it will be seen that the cylinder is open to the annular space formed round the piston valve stem 9, and between the piston ports of the valve. These form the combustion chamber, and the explosive mixture is compressed into them and ignited by the tube igniter 10. When the piston 1 has made its power stroke down, the piston valve moves to bring into connection the ports 11 and 12, and the piston 1 then moves up and discharges the exhaust products; on the next stroke down the piston valve again takes the position shown by the upper valve, and the lower valve 13 is opened to admit a charge of gas and air on the next down stroke. The piston 2, as shown on the drawing, is just finishing its exhausting stroke, and the piston valve is about to close the exhaust port. The valve arrangements of piston 2 are similar to those of piston 1.



Top cylinder, 400 revs. per min.



Bottom cylinder, 480 revs. per min.

FIG. 106.—Diagrams from Top and Bottom Cylinders, Burt's High Speed Engine.

'This engine is most interesting for many reasons; its designers are very daring, and appear to the author to disregard some of the understood conditions of gas engine economy. It appears to him impossible to obtain any high economy in gas consumption from an engine with its combustion spaces made up of tortuous ports and passages. The engine gives the designer an extreme example in the direction of subdividing the combustion space, and it will certainly be interesting to know its power and gas consumption. Fig. 106 gives diagrams taken from the top and bottom cylinders at 400 and 480 revolutions respectively.'

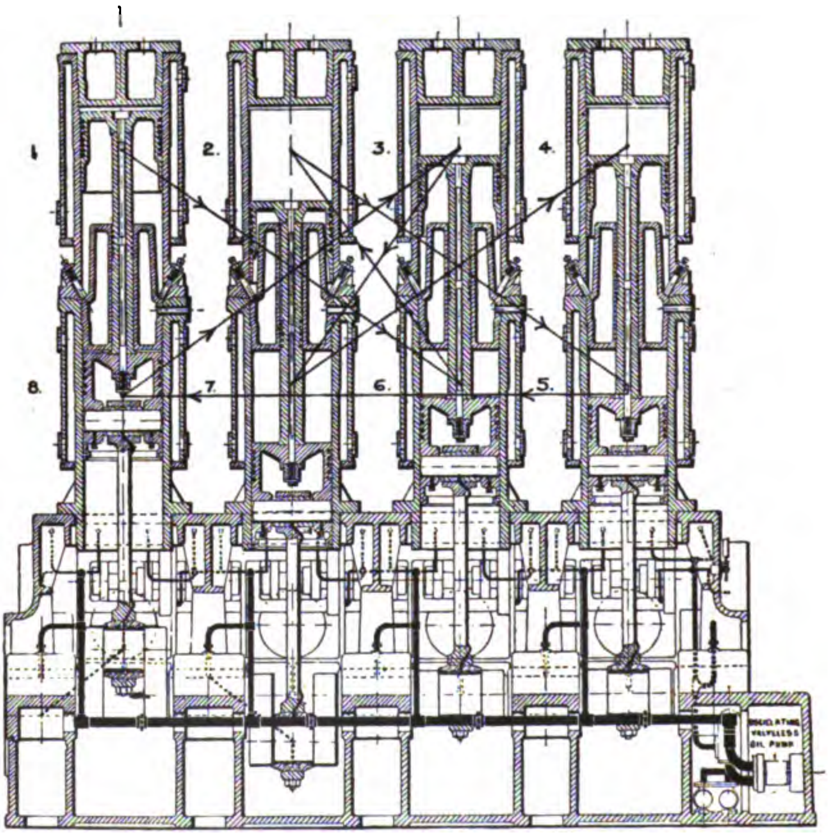


FIG. 107

The tandem vertical engine of the Westinghouse Co. follows this type, but without using piston valves; they use the ordinary lift valve both for inlet and exhaust, and the combustion space in the relatively large engine cylinders adopted has reasonable proportions as regards the relation of cubic capacity to surface.

Fig. 107 is a section of a Westinghouse engine of 1000 HP, for which the author is indebted to Mr. Allen's interesting paper on large gas engines.[1]

Fig. 108 is a diagram of the mode of firing. Fig. 109 is an external view of a 750 HP engine, for which the author is indebted to the English Westinghouse Co.

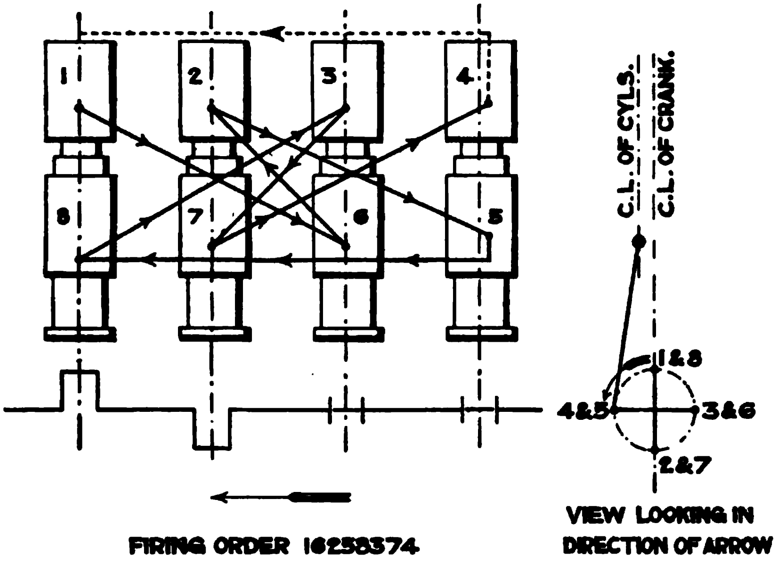


FIG. 108

Mr. Percy Allen in his paper describes the Westinghouse practice as follows:

'It has already been mentioned that the American Westinghouse Machine Company brought out some years ago a vertical, single-acting, three-crank gas engine using the Otto cycle, and the English company developed the idea by duplicating the cylinders and making the trunk pistons of two cylinders act on the same crank. A complete line of engines of this type has been designed, the diameter of the cylinders being kept uniform, and the stroke and speed being the same throughout. Each line of cylinders represents a single unit of power, increased output being obtained by simply multiplying the number of lines of cylinders. In these engines the stroke is in all cases 24 ins., the speed 200 revolutions per minute or thereabouts, and the diameter of the lower cylinder 21 ins. and the diameter of the upper 22 ins. This difference allows the lower trunk to be readily drawn up and removed through the top cylinder. One line of tandem cylinders of these dimensions represents 250 BHP, two lines represent 500 HP, three lines 750 HP, and a four-crank engine with four lines or eight cylinders equals 1000 HP. The upper and lower cylinders are separated by means of an intermediate head, which has a conical projection extending up into the top cylinder, so as to obtain sufficient length to form a stuffing-box. The upper and lower trunk pistons are separated by a cast-iron distance piece and held together by a nickel steel bolt, and the stuffing-box action is obtained by a series of piston rings carried in the distance-piece. The upper part of the connecting-rod works in a gudgeon pin in the lower trunk in the usual way. The bottom of the upper cylinder being closed in by the top of the intermediate piece, forms an air-buffer to the top trunk, the pressure thus produced being so proportioned that it absorbs the inertia of the reciprocating parts; and that it has this effect is proved by the fact that these engines run very steadily without any counterweights on the cranks. The cylinders, top covers, and intermediate heads are water-cooled; but it has not been found necessary to water-cool either the trunks or the exhaust valves.

---

[1] 'Large Gas Engines,' by Percy R. Allen, *Cassier's Magazine*, July, August, and September, 1909.

'In the 1000 HP, four-crank size, each adjacent pair of cylinders have the cranks opposite one another, but the two pairs of cranks are at right angles. The diagrams, figs. 107 and 108, explain this, and also explain the sequence of firing in the various cylinders. There is only one inlet valve and one exhaust valve to each cylinder, and these are worked in the simplest possible way from a 2–1 cam shaft placed in the enclosed bed of the engine.

'Forced lubrication is obtained from two valveless pumps worked off the crankshaft, and is used for the lubrication of the main bearings, cam shaft bearings, and both ends of the connecting-rod. The cylinder lubrication is effected by separate pumps with sight-feed lubricators. The governing is effected by throttling both the air and the gas.

'So far high-tension ignition has been used on these engines, but the revision of the ignition question is now under consideration.

'The writer had four of these engines, each of 750 HP, working for some time very satisfactorily; but when it came to put down the 1000 HP engine with eight cylinders it was found that the plan of making all eight cylinders draw their supply of gas through one manifold did not answer, as the groups of cylinders most favourably situated with regard to the gas and air supply did the bulk of the work while the others were starved. This was overcome by dividing the cylinders into groups of four each and connecting them quite independently to the gas main and air inlets; that is to say, as far as the supply of mixture is concerned the engine might be regarded as being divided into two sections with the crankshaft common to both. It is a very debatable point as to how far this type of engine is suitable for large powers. Up to eight cylinders it seems a manageable machine, and has much to commend it in cases where floor space is valuable. Comparatively inexpensive foundations are required, and none of the individual parts is of unwieldy size. If the

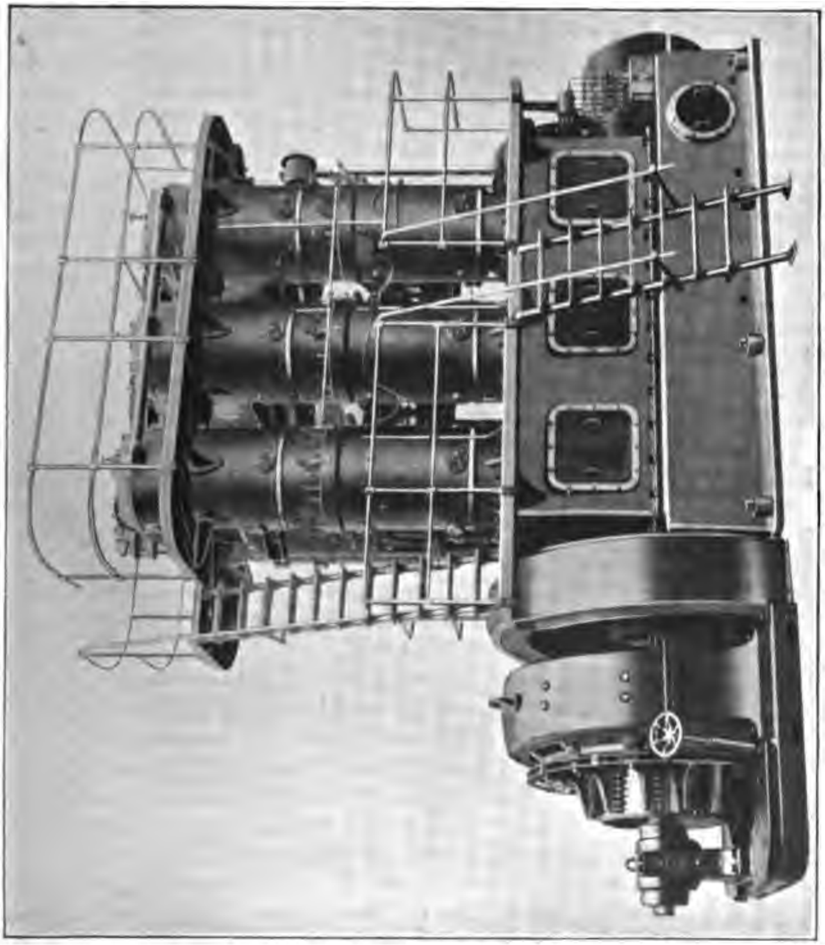


Fig. 109

ignition is kept in good order it gives very even turning; that it has been successfully applied to cotton spinning proves this.

'The consumption of gas per BHP is certainly as low as any other type of engine, and the lubricating arrangements require only a very moderate amount of oil.

'On the other hand, the design involves in the larger sizes a multiplication of cylinders and a duplication of a number of small working parts, which, although simple enough individually, when combined present quite a number of points for possible breakdowns; and in the present state of the art it seems probable that, for land purposes and for powers above 1200 HP, the tandem horizontal type of engine will, on the whole, be preferred. The writer has specially referred to the Westinghouse engine, as up to now this has been the only vertical design in which 1000 HP has been reached; but at the same time there are numerous other designs of vertical gas engines of smaller sizes operating very successfully in various countries: amongst others, the vertical gas engine of Crossley, Fielding & Platt, Tangye, and Campbell will be frequently met with, while on the Continent the Guldner vertical engine has come into considerable use, and in the United States there are a number of makers who build three and four cylinder vertical engines up to moderate sizes.'

The Westinghouse tandem vertical engine of 1000 HP and less has attained considerable success in Britain, and the difficulties of distribution to the various cylinders have been completely overcome. It is true that such engines necessarily have a considerable number of parts, but reference to the insurance statistics proves conclusively that gas engines of moderate dimensions are if anything more reliable than steam engines, so that certainly up to 1000 HP such engines are fully equal to large cylinder gas engines from this standpoint.

The directors of the National Gas Engine Co. have also considered the multiple cylinder with unwatered piston as a safe solution of the large power gas engine problem, and they have put upon the market tandem single acting vertical engines of power from 185 to 1500 BHP. For this purpose they have built a new works devoted solely to the construction and testing of tandem vertical engines.

Fig. 110 shows an external view of a three-crank, six-cylinder vertical engine rated at 750 BHP. The upper cylinders are 23 ins. diameter and the lower 22 ins. diameter; the stroke is 24 ins. Each upper and lower piston is connected by a cast-iron sleeve of 6 ins. diameter; through this passes a nickel steel bolt of high tensile strength, which compresses the two pistons against its ends. The pistons present a smooth unbroken surface to the gases of the explosion in order to avoid heated parts likely to cause pre-ignition. They are also of the special construction used by the National

Company for all their larger cylinders. An inner strengthening ring is cast from near the centre of the bottom of the piston to the cylindrical

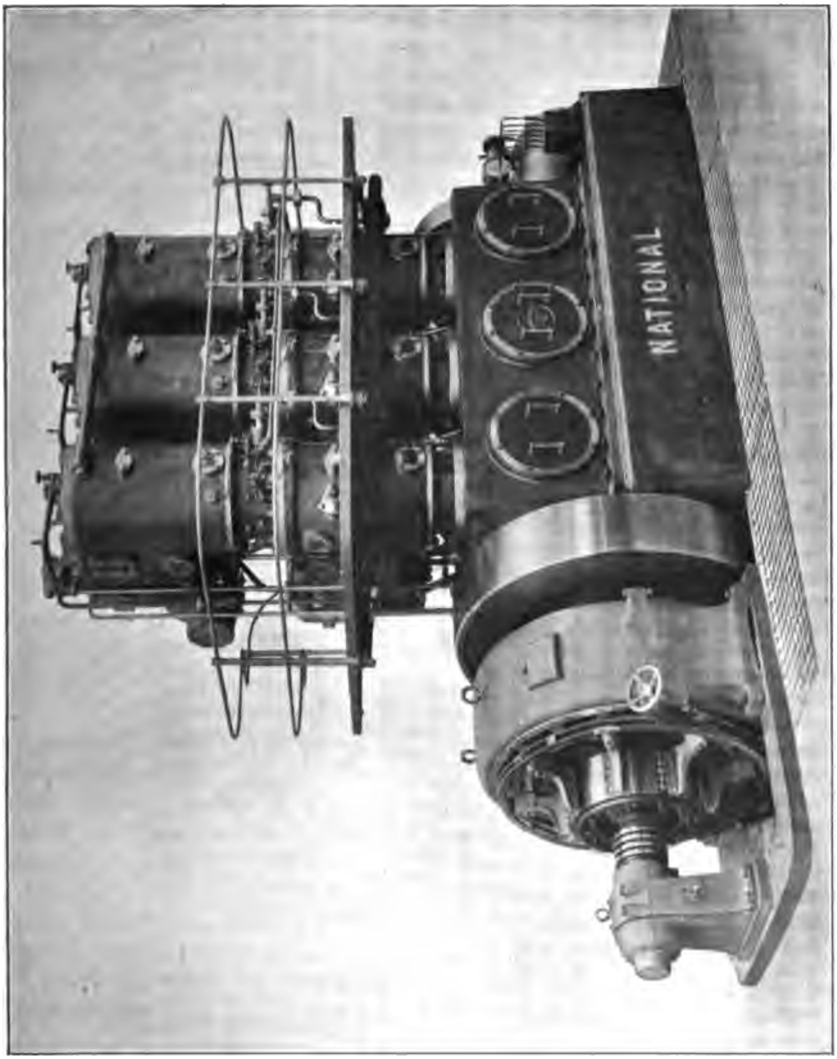


FIG. 110

side at some distance from the bottom (*v.* fig. 56). The centres of all unwatered pistons tend to heat up because of the slow rate of conductivity through a long cast-iron path ; by this contrivance two metallic paths are provided for the heat flow from the centre of the bottom, and the practical result is that even under great overloads these pistons do not heat up to any temperature which can cause pre-ignition. The margin of safety is so considerable that very probably 26 ins. diameter pistons may be safely run in such engines without water.

The inner strengthening cast-iron ring has openings to allow of the free circulation of air up to the piston bottom.

The cast-iron sleeves are not fitted with piston rings as was done by Burt and also by Westinghouse; they are made pressure tight by a simple metallic packing fitted into the upper part of the water-jacketed distance piece through which the sleeve or piston rod slides. A section of this packing is shown at fig. 111. It consists simply of special cast-iron springs, accurately turned and cut to spring inwards and grip the sleeve; these rings are porportionately very deep, so as to allow metal for long continued wear; they are separated by L rings scraped together, and held tight by a cover and bolts. The cast-iron sleeve is not water cooled in any way, but its contact with the interior of the watered distance piece or middle cover keeps its temperature so low that the part rubbing on the rings takes a high and permanent polish. This packing is at once simple and successful.

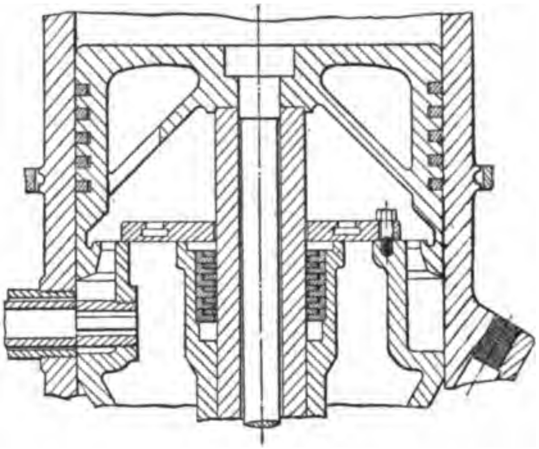


FIG. 111

The lower side of the upper piston acts with the middle cover as an air cushion in which air is alternately compressed and expanded, and the cushion greatly aids smooth running by assisting to reverse the pistons at the bottom ends of their stroke. The cranks are carefully balanced to further ensure smooth running.

It will be noted that the two cylinders are cast separately, and their water jacket casings are cast with the liners but with a gap which leaves casing and liner free to expand independently. A packing ring is arranged to retain the water without interfering with freedom of relative movement.

The lower walls of the cylinders are not water jacketed, and here are provided the inclined screw pin fastenings for the middle cover and the bolt and flange fastenings for the lower cylinder. The middle cover seats in a metallic joint, and the joints of the piston sleeve top and bottom are both metal to metal.

Forced lubrication is applied to all reciprocating and revolving parts of the engine at an oil pressure of about 20 lbs. per sq. in.; after discharge from the bearings the oil returns to the crank casing and flows to the governor end of the bed, where it is filtered before passing into the valveless pumps from which it is again forced through the engine. The bottom cylinders are lubricated from the oil leaving the crosshead pin, and the upper cylinders and metallic packing are supplied from a pump with sight feed operated from the engine.

The inlet and exhaust valves open into a port common to both and are placed one above the other. The combustion space formed has a larger proportion of surface exposed for a given cubic content than is usual in horizontal engines of the smaller type, although the ratio is quite as favourable as that of many larger horizontal double-acting engines in which ports or pockets are provided for the valves above and below the cylinders.

Inlet and exhaust valves are of cast iron without watering, and the seats are also cast iron removable to allow of renewal. The arrangements of valve shaft, two to one gear wheels, and levers and links are plainly seen in the section; the cams and rollers are of steel, case-hardened. The inlet valve carries a gas valve on its stem, which valve does not seat but works closely within a bored-out cylindrical passage. The gas and air passages are formed by a partitioned casing and pipes, and the gas is admitted later and closed earlier than the air for the reasons already discussed in other engines. Governing is accomplished by the quantity method, in which the charge is kept of constant composition but varied in density.

Compressed air is used for starting.

Ignition is accomplished by two high-tension plugs to each cylinder, with Bosch magneto.

A test of one of these engines was made at the National Co.'s works, Ashton-under-Lyne, by Mr. Wm. Stead, M.I.Mech.E., M.I.E.E., who is mainly responsible for the design.

The results are given on p. 172.

TEST OF A NATIONAL VERTICAL TANDEM GAS ENGINE RATED AT 750 BHP. (*Stead*)

Particulars of engine: Three crank tandem vertical engine; six cylinders: three upper cylinders, 23 ins. dia., three lower, 22 ins. dia.; piston sleeve, 6 ins. dia.; stroke, 24 ins.

Engine No. 251

Load	Overload	Full Load	¾ Load	½ Load	¼ Load	No Load
Volts	260	243·2	243·2	242·5	256	—
Amps.	2439·2	2121	1602·5	1040	540	—
KW	621·97	515·0	389·0	252	138	—
BHP	903	748	571·0	380	208	—
Time	11 to 12 a.m.	2 to 3 p.m.	3·15 to 4 p.m.	4·15 to 4·45 p.m.	4·50 to 5·5 p.m.	5·7 to 5·25 p.m.
Length of test	60 mins.	60 mins.	45 mins.	30 mins.	15 mins.	18 mins.
Barometer	30·43″	30·43″	30·43″	30·43″	30·43″	30·43″
Water gauge (gas pressure)	2″	2″	2″	2″	2″	2″
Gas used	62,000 cb. ft.	49,900 cb. ft.	31,200 cb. ft.	16,300 cb. ft.	6480 cb. ft.	4870 cb. ft.
Gas used per hour	62,000 " "	49,900 " "	41,600 " "	32,600 " "	25,920 " "	16,200 " "
Gas used per KW hour	99·6	96·9	107	129·2	187·8	—
Gas used per BHP hour	68·7	66·6	72·9	85·7	124·6	—
Gas temperature	63° F.	71° F.	73° F.	72° F.	72° F.	—
Gas used per KW, red. to 60° F., 30″ Hg.	99	96·6	106·2	128·5	186·6	—
Gas used per BHP	68·4	66·4	72·3	85	124	—
Calorimeter:						
Gas value, higher val.	164	156·25	138·5	158·6	158·6	—
" lower "	151·5	146·75	129·2	148·5	148·5	—
When 60° F., 30″ Hg, higher val.	163·3	157·25	139·5	159·9	159·9	—
" " lower "	151	148·6	139·15	149·8	149·8	—
B.Th.U. used	9,400,000	7,280,000	4,030,000	2,420,000	963,000	—
B.Th.U. per KW hour	15,070	14,350	13,820	19,240	28,000	—
B.Th.U. per BHP "	10,400	9860	9420	12,720	18,580	—
Cooling water:						
Gallons used per hour	6300	4900	4900	4,300	4,000	—
Gallons used per BHP hour	6·96	6·55	8·6	11·3	19·25	—
Temp. of water inlet	57° F.	58° F.	58° F.	59° F.	59° F.	—
" " outlet	—	100° F.	—	—	—	—
Vacuum in inlet pipe, inches Hg	2″	5½″	8½″	13½″	15″–16″	17″
Thermal efficiency per BHP, per cent.	24·45	25·8	27	20	13·75	—
Revolutions per minute	203	204	204	206	206	206

The gas was measured by a Thorp and Marsh rotary anemometer type meter and the load was taken by dynamo. All instruments were carefully calibrated.

From these particulars it will be seen that the engine though rated at 750 BHP readily carried an overload of 903 BHP. The brake thermal efficiency at full load, 748 BHP, was 25·8 per cent., and at ¾ load, 571 BHP, 27 per cent.; even at half load the brake thermal efficiency was still 20 per cent.

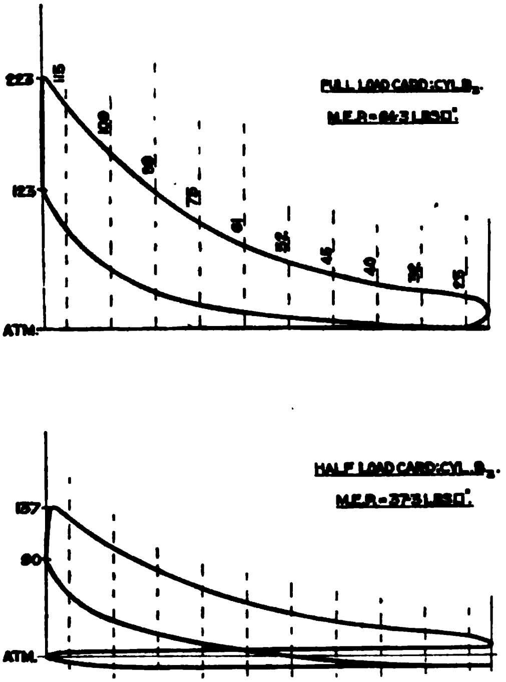


FIG. 112

The mechanical efficiency of this engine was found to be 90 per cent., giving an indicated thermal efficiency at ¾ load of 30 per cent.

The results are quite satisfactory for an engine in ordinary working condition without any attempt at the best possible adjustment of gas and air for maximum economy. It is quite evident that the engine could have been adjusted for from 35 per cent. to 37 per cent. indicated thermal efficiency.

The National Co., like other large gas engine builders, guarantee the brake thermal efficiency to be not less than 25 per cent. or 10,000 B.Th.U. per BHP hour; but in ordinary use much better results are obtained, and by careful adjustment the highest test results can be got.

Indicator cards were taken by Mr. Stead during his test; of which

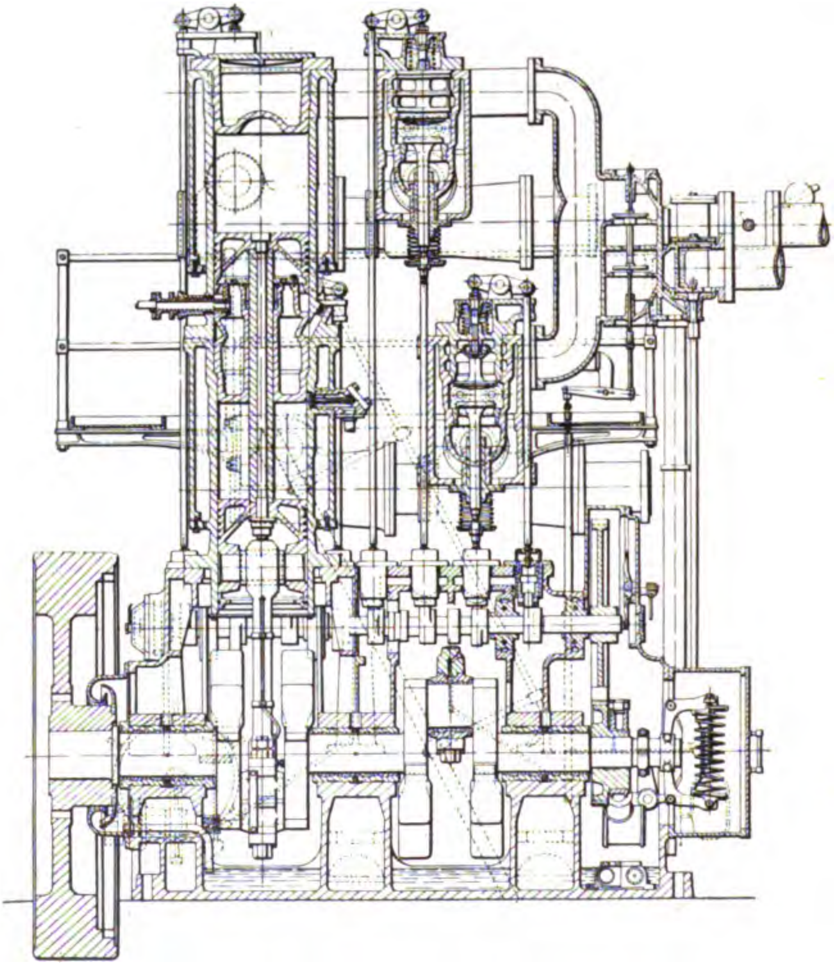


FIG. 113 *a*.

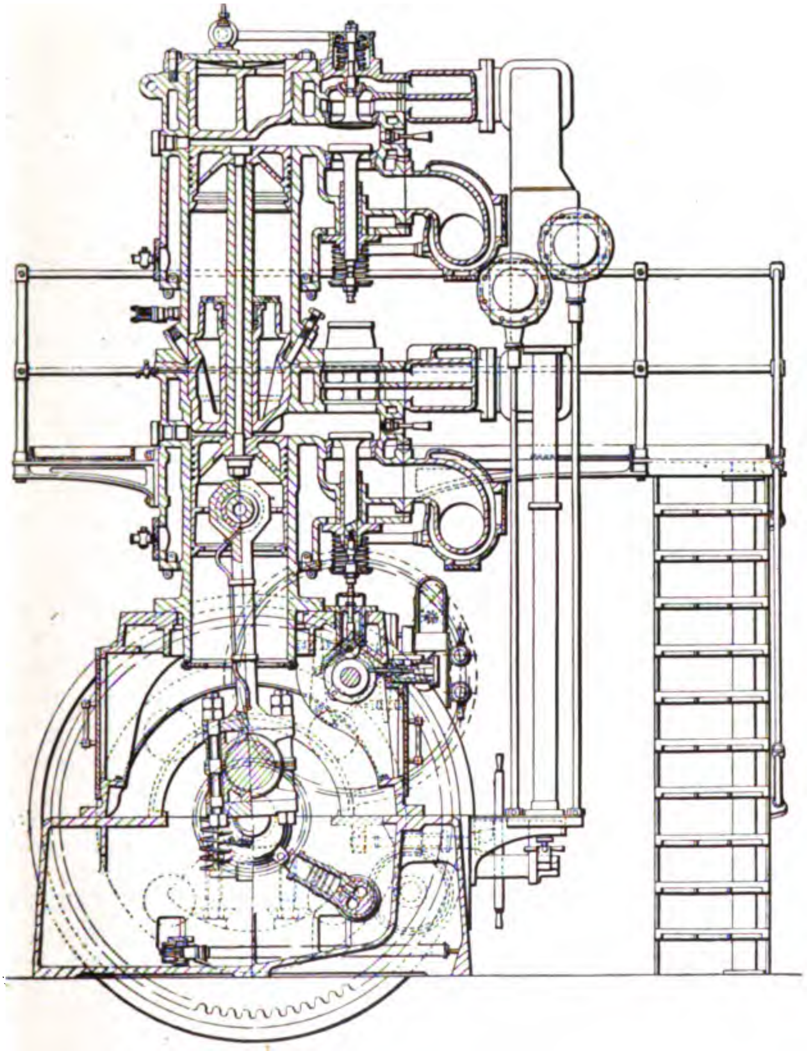


FIG. 113 *b*.

fig. 112 shows full load and half load cards of cylinder B^3, the third of the bottom cylinders.

The ratio of compression $\frac{1}{r} = \frac{1}{6}$.

From the card it appears that the compression pressure at full load is 123 lbs. per sq. in. above atmosphere, and the explosion pressure 223 lbs. above atmosphere. The mean pressure is 64·3 lbs. per sq. in.

At half load the compression pressure is 90 and explosion pressure 137 lbs. per sq. in. above atmosphere; the mean pressure is 37·3 per sq. in. The ratio of mean to maximum pressure is thus about 3·5.

At the overload of 903 BHP the compression pressure was 165 lbs. per sq. in. above atmosphere, and the mean pressure 73·7 lbs. per sq. in.

The table on p. 178 gives various particulars of the engines of this vertical tandem type now built by the National Co.

Figs. 113 *a* and *b* show longitudinal and transverse sections of a two-crank engine of 300 BHP with four cylinders; the upper cylinders are 18 ins. diameter and the lower 17 ins., the piston sleeve diameter is 4½ ins., and stroke 18 ins. The revolutions per minute are 300.

A recent brake test of this engine made at the works showed 362 BHP and 404 IHP at 299 revolutions per minute with a mechanical efficiency of 89·6 per cent. A Heenan and Froude water brake was used in the test. The engine took an overload of 360 BHP continuously when required.

Fig. 114 is from a photograph of two four-crank engines having each eight cylinders, and one three-crank six-cylindered engine, which ran at the Crystal Palace during the Festival of Empire Exhibition for six months in 1911 on coal gas supplied by the South Suburban Gas Company, generating electricity for lighting and power purposes. The two eight-cylindered engines have the same bore and stroke as the engine shown in fig. 110, and each develops 1000 BHP at 200 revs. per min. when run on anthracite producer gas at 60 lbs. per sq. in. mean pressure on the pistons; an overload to 1100 BHP is sustained with ease. The six-cylindered engine is of 750 BHP.

These engines are now in successful operation, and although the special works erected by the National Co. for their manufacture has only been at work for one year, the engines delivered and now under construction amount to 20,000 BHP.

At present three 1500 BHP engines are under construction for Japan; they have each six cranks and twelve cylinders of 24 ins. stroke; the upper cylinders are 23 ins. diameter and the lower 22 ins.

It is probable that the Company will build cylinders of 26 ins. and 25 ins. diameter for four and six crank units which will give 1300 and 2000 BHP as rated loads respectively. In our present state of knowledge it would be inadvisable to attempt larger cylinders without watered pistons.

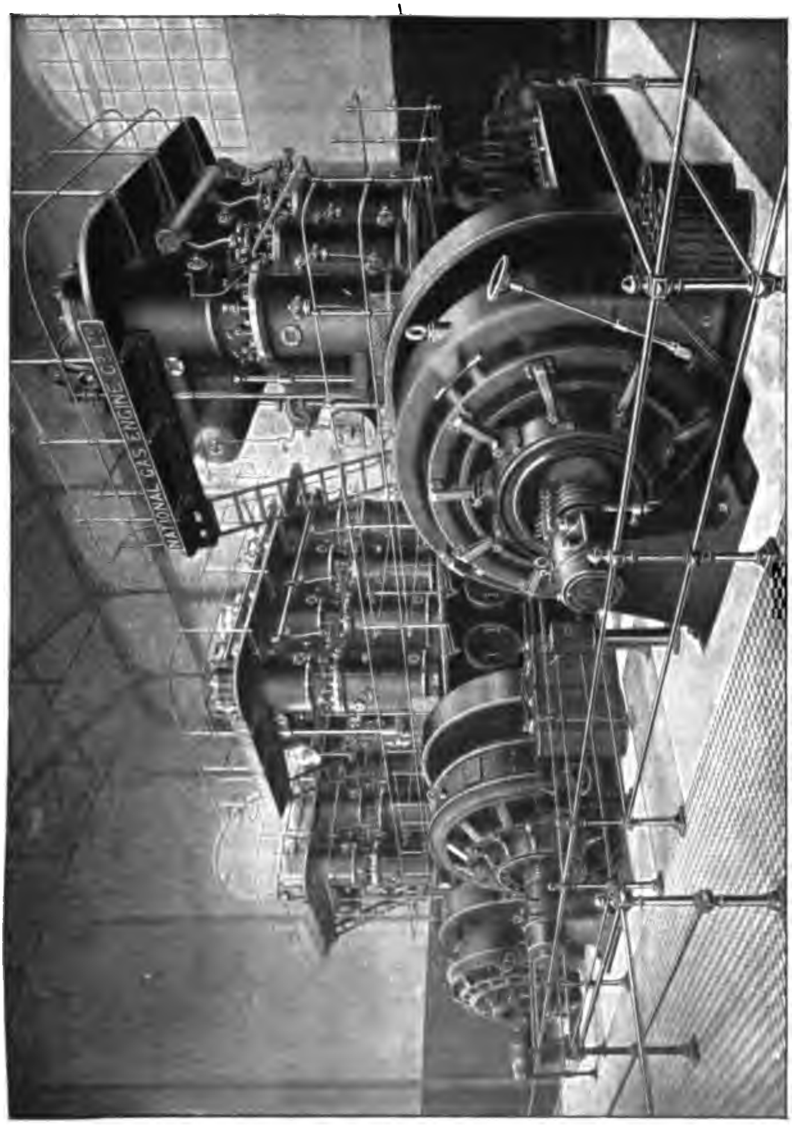


FIG. 114

VERTICAL TANDEM GAS ENGINES

POWER OF ENGINES

National Gas Engine Co., Ltd.

Particulars of engine				Coal gas. 550 B.Th.U. per cub. ft.				Coke oven gas. 400–450 B.Th.U. per cub. ft.				Producer gas from anthracite. 135–140 B.Th.U. per cub. ft.				Producer gas from coke. 120 B.Th.U. per cub. ft.				Blast furnace gas. 95–100 B.Th.U. per cub. ft.			
Stroke	No. of cranks	Diameter of cylinders	Revs. per minute	MEP	Rated BHP	10 per cent. overload BHP	Rated KW 92 per cent. gen. eff.	MEP	Rated BHP	10 per cent. overload BHP	Rated KW 92 per cent. gen. eff.	MEP	Rated BHP	10 per cent. overload BHP	Rated KW 92 per cent. gen. eff.	MEP	Rated BHP	10 per cent. overload BHP	Rated KW 92 per cent. gen. eff.	MEP	Rated BHP	10 per cent. overload BHP	Rated KW 92 per cent. gen. eff.
18″	1	17″ & 18″	300	60	185	205	128	55	170	187	117	48	150	165	102	455	142	156	97	41 41	125 150	137 165	85 102
18″	2	17″ & 18″	300	60	375	410	257	55	340	374	233	48	300	330	205	455	285	314	195	41 41	250 300	275 330	171 205
18″	3	17″ & 18″	300	60	560	615	385	55	510	561	355	48	450	495	307	455	427	470	293	41 41	375 450	412 495	257 307
18″	4	17″ & 18″	300	60	750	825	515	55	680	748	466	48	600	660	410	455	570	618	390	41 41	500 600	550 660	343 410
24″	3	22″ & 23″	200									54	750	825	515	51	665	730	487	45 46	630 750	693 825	730 515
24″	4	22″ & 23″	200									54	1000	1100	687	51	950	1045	650	45 46	840	925	575

COMPARISON OF THE DIFFERENT TYPES OF FOUR-CYCLE ENGINES

It will be seen from the foregoing account of the development of these engines, that the smaller engines up to 200–250 BHP are built preferably of the single-acting open-cylinder type as single horizontal cylinders, while for large engines the favourite type is the horizontal single-crank tandem double-acting engine, in which the two cylinders between them give the crank two impulses per revolution. These tandem engines are built from 22 ins. to 52 ins. cylinder diameter, giving from 400 to 3000 BHP to one crank.

Vertical engines are also considerably used, especially in England; and for power up to 2000 BHP per four crank unit in the tandem single-acting type they present considerable advantages which make them formidable competitors with the tandem horizontal double-acting engine. A certain number of single-acting tandem horizontal engines have also won a position in the market, but on the whole opinion tends to look at the others more favourably.

A comparison of the weights per BHP of different types of engines is useful as showing the reasons for preferring the multiple and double-acting cylinders for large gas engines. But first it is to be remembered that in all engines the weight per unit of power developed increases with increasing dimensions. The following table shows the total weight and the weight per BHP developed for various types of engines. The first part of the table gives particulars of National gas engines of the horizontal type, each engine having one cylinder of the open trunk single acting type. The engines are those of the latest design as now built by the company, and illustrated at fig. 52.

Particulars of the large single-cylinder Cockerill engine are also given, and also two examples of the favourite Continental horizontal tandem double-acting type, and lastly two examples of the new National tandem single-acting vertical engines as shown at fig. 110.

SOME PARTICULARS OF GAS ENGINES OF DIFFERENT DIMENSIONS AND TYPES

HORIZONTAL NATIONAL GAS ENGINES, SINGLE CYLINDER, SINGLE ACTING, FOUR CYCLE

Rated BHP with coal gas	Dia. and Stroke	Stroke bore ratio	Revolutions per min.	Piston speed, ft. per min.	Wt. of engine, tons	Wt. per rated BHP
57	13″ × 20″	1·538	230	498	3·33	131 lbs.
68	14″ × 21″	1·500	230	537	4·08	134·5 lbs.
85	16″ × 22″	1·375	210	560	5·82	153·5 lbs.
105	17″ × 26″	1·530	190	539	9·08	193·5 lbs.
125	18·5″ × 28″	1·513	180	555	9·48	170·3 lbs.
150	20″ × 31″	1·550	170	567	13·20	197·7 lbs.
185	22″ × 34″	1·545	160	587	17·05	206·5 lbs.

Horizontal Cockerill Single Cylinder, Single Acting, Four Cycle.

Rated BHP with blast furnace gas.	Dia. & stroke.	Stroke bore ratio.	Revs. per min.	Piston speed, ft. per min.	Wt. of engine, tons.	Wt. per rated BHP.
800	51½″ × 55″	1·07	75	470	160	448 lbs.

Horizontal Continental Tandem, Double Acting, Four Cycle.

Rated BHP with blast furnace gas.	Dia. & stroke.	Stroke bore ratio.	Revs. per min.	Piston speed, ft. per min.	Wt. of engine, tons.	Wt. per rated BHP.
800	28″ × 34″	1·21	130	736	85	238 lbs.
1000	32″ × 40″	1·25	110	733	120	269 lbs.

Vertical National Gas Engines, Tandem, Single Acting, Four Cycle.

Rated BHP with blast furnace gas.	Dia. & stroke.	Stroke bore ratio.	Revs. per min.	Piston speed, ft. per min.	Wt. of engine, tons.	Wt. per rated BHP.
750	18″ × 18″	1·00	300	900	39·0	116·5 lbs.
1250	26″ × 24″	1·00	200	800	83·5	149·0 lbs.

It will be observed that while the horizontal National gas engine of 57 BHP with a 13 in. diameter cylinder gives 1 BHP for 131 lbs., the 185 BHP requires 206·5 lbs.

The large Cockerill horizontal engine requires 448 lbs. weight for each BHP. The tandem double-acting engines require respectively 238 and 269 lbs. per BHP, and the two National vertical single-acting tandem 116·5 and 149 lbs. per BHP respectively.

For each separate type there is an increase of weight per BHP with increasing power or cylinder diameter.

This is more clearly shown at fig. 115, where the weights per BHP are plotted against cylinder diameter.

In the open trunk single-acting horizontal engines the rate of increase of weight with increasing cylinder diameter appears to be linear. Six out of the seven National engines fall closely round a straight line drawn from the point indicating the weight of the large open trunk Cockerill engine. The one weight which is off the line is that of the 17 in. cylinder, in which the frame was purposely made heavy in order to save patterns by using it also for the next size of 18·5 in. cylinder. This line then shows that while a 9 in. cylinder would involve about 100 lbs. per BHP, a 56 in. cylinder would require about 500 lbs. per BHP.

The vertical National engines show that a 9 in. cylinder would require less than 60 lbs., and a 30 in. cylinder under 200 lbs. per BHP; a 30 in. cylinder of the open trunk type would require just under 280 lbs. per BHP.

The weights for the Continental tandem double-acting engines come between the upper and lower lines.

The following table gives the weights per BHP of nine Continental engines of the horizontal tandem double-acting type, from which it will be seen that there is a fairly regular increase of weight per BHP with increasing power.

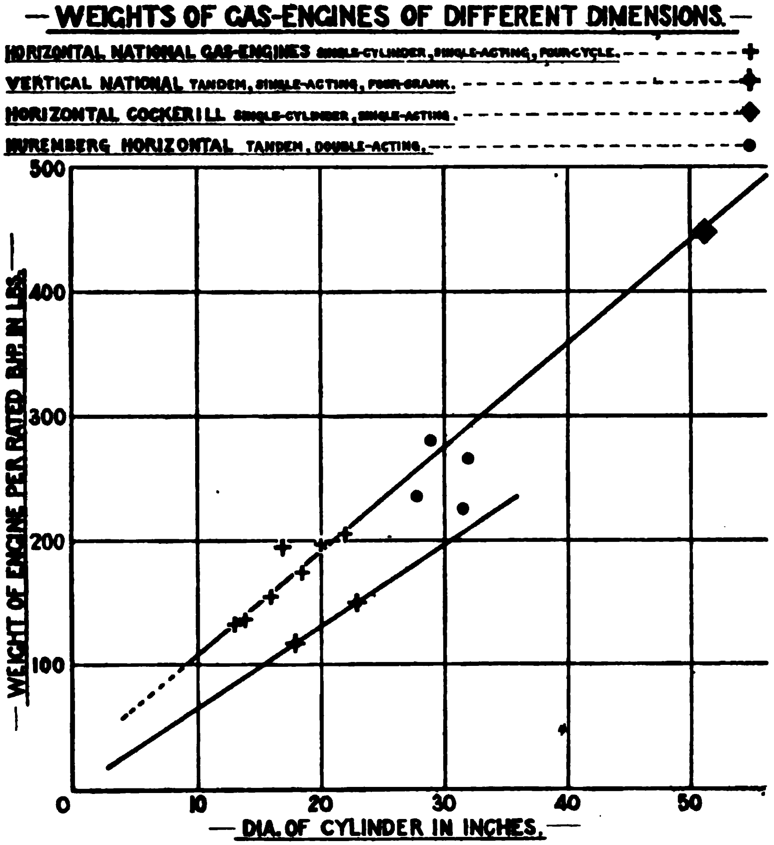


Fig. 115

WEIGHTS OF LARGE CONTINENTAL GAS ENGINES OF THE TANDEM, DOUBLE-ACTING, FOUR-CYCLE TYPE

Brake HP.	Nett weight of engine with fittings but without flywheel.	Nett weight of flywheel.	Weight of engine per BHP.	Weight of flywheel per BHP.
2000	266·5 tons	—	298·5 lbs.	—
1500	162·3 tons	34·5 tons	242·5 lbs.	51·3 lbs.
1060	109·3 tons	—	231·0 lbs.	—
1000	120·0 tons	—	269·0 lbs.	—
950	95·1 tons	25·0 tons	224·0 lbs.	58·8 lbs.
800	85·0 tons	—	238·0 lbs.	—
750	68·5 tons	17·7 tons	204·5 lbs.	52·6 lbs.
700	81·7 tons	15·7 tons	261·0 lbs.	50·2 lbs.
630	66·5 tons	—	237·0 lbs.	—

Fig. 116 shows the weight of the engine per BHP plotted against rated BHP for the engines of which some particulars are given in the two tables.

The horizontal single-acting engine has obviously the most rapid rate of increase when compared either with the tandem double-acting or the vertical tandem single-acting.

On line A a weight of 300 lbs. per BHP only gives an engine of about 470 BHP, while on B a power of 2050 BHP is obtained for the same weight; this same power of 2050 BHP, however, can be obtained on line C by a weight of 200 lbs. per BHP.

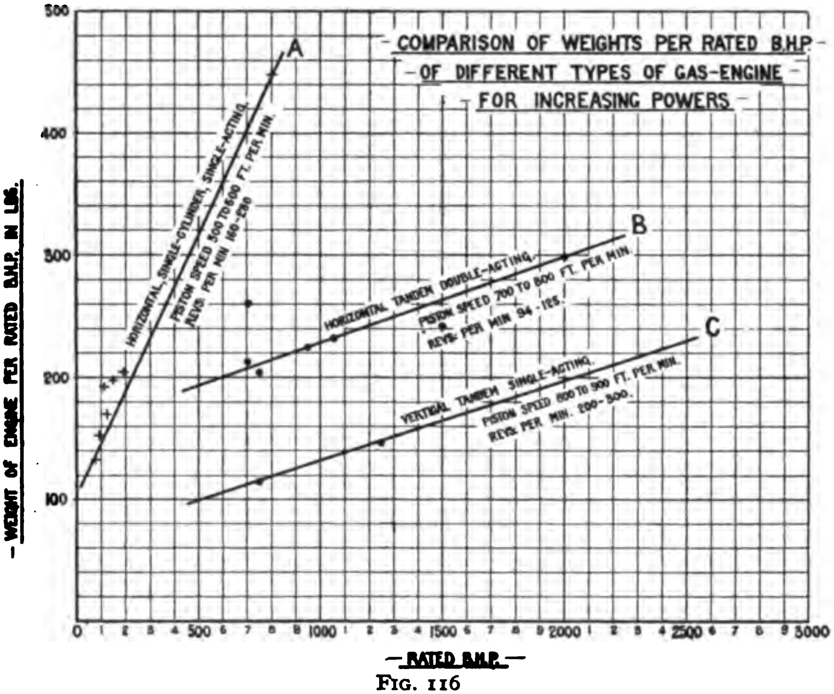


FIG. 116

The diagram very clearly shows the reasons of the success of the Continental horizontal tandem and also the reasons which induce the English designers of the Westinghouse and National tandem vertical engines to produce multiple cylinder engines for large powers.

The various points of interest in large and small four-cycle engines have now been shortly discussed, and this chapter may be fittingly closed with the following table giving some particulars of Continental horizontal tandem double-acting engines as published by Messrs. Haniel & Lueg of Dusseldorf, who rank high among the engine builders of Germany.

Fig. 117 shows a diagrammatic longitudinal elevation and plan of the engines referred to in the following table:

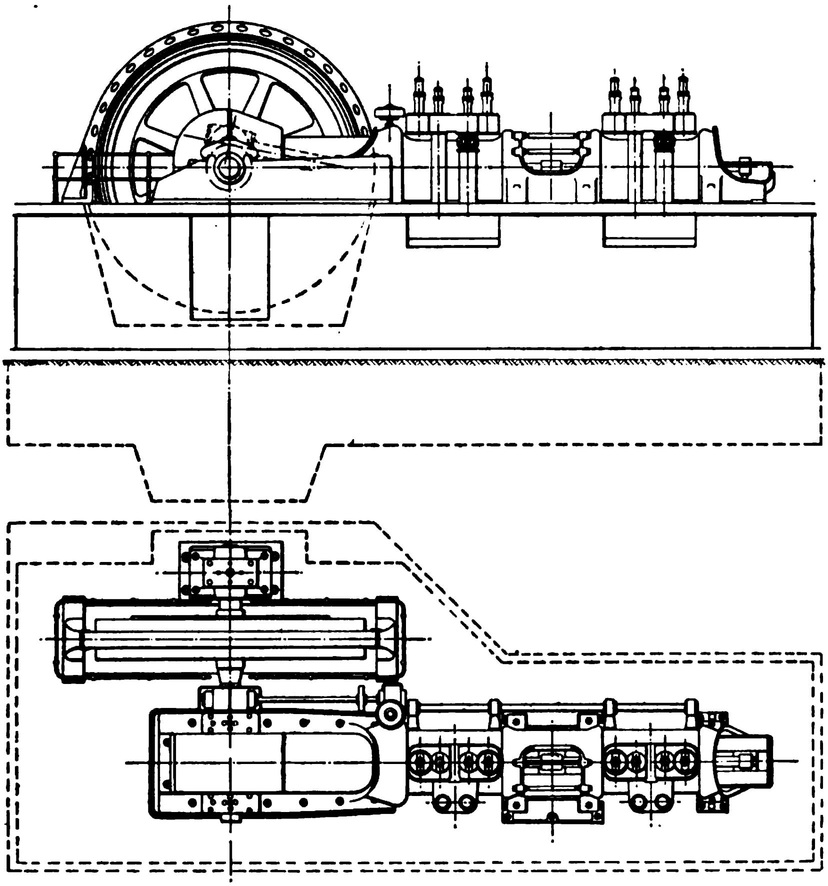


FIG. 117

PRINCIPAL DIMENSIONS AND CAPACITIES OF THE HORIZONTAL TANDEM DOUBLE-ACTING ENGINES BUILT BY MESSRS. HANIEL AND LUEG OF DUSSELDORF

Capacity actual BHP.	Dia. and Stroke.	Revolutions per min.	Piston speed, ft. per min.	Total length.
350	20·5″ × 25·56″	150	640	31·2 ft.
500	23·62″ × 27·5″	150	687	33·4 ft.
620	25·62″ × 29·5″	150	738	35·8 ft.
700	27·93″ × 31·5″	125	657	38·0 ft.
900	30·69″ × 35·44″	125	739	42·0 ft.
1050	33·87″ × 39·37″	110	722	47·0 ft.
1480	38·81″ × 43·31″	107	722	50·5 ft.
1700	41·37″ × 47·25″	94	740	56·0 ft.
2400	47·25″ × 51·187″	94	803	62·0 ft.
3000	50·44″ × 51·187″	94	803	66·0 ft.

CHAPTER II

THE DEVELOPMENT OF TWO-CYCLE GAS ENGINES

FROM a mechanical point of view the Otto cycle is a very imperfect one, involving as it does, at most, only one impulse for every two revolutions of the crankshaft for each single-acting power cylinder. The mechanical imperfections of the engine up till now, however, have been more than compensated for by the simplicity due to the single cylinder and piston, and the readiness with which the practical conditions of charging, exploding, expanding, and exhausting can be made to comply with the requirements of economical working.

At first it seems to many engineers a simple matter to design a compression gas engine capable of giving an impulse at every revolution for each single-acting cylinder. The problem is in reality one of the most difficult in the whole range of engineering, and, although it has been the subject of much continuous effort, no solution has yet been found which completely satisfies simultaneously the commercial conditions of the problem for both large and small engines.

It has always been observed that before a satisfactory solution of any important engineering problem is attained, many proposals or attempts are made by acute inventors; and, although some of them attain partial success, yet, as a rule, years elapse, one inventor succeeding another, and carrying the subject a little further, before sufficient knowledge is accumulated to make the matter commercially successful. The gas engine has a history of this kind, which has been briefly sketched in the first volume.

It may be said here, however, that in 1862 a French engineer actually proposed the Otto cycle gas engine and discussed carefully the conditions of its success, but left it to be independently discovered and reduced to practice by Otto in 1876. Otto began his long struggle with gas-engine difficulties in 1854, and achieved a partial success in 1867 with the Otto & Langen atmospheric engine, which itself was first proposed by Barsanti and Matteucci in 1857. He worked and studied until, in 1876, he invented the famous Otto engine which has proved the turning-point in gas-motors. He had thus laboured at his subject for twenty-two years before real success crowned his efforts.

It has often occurred to the author how much more rapidly engineering science would advance if inventors placed on record some description of their experimental failures as well as accounts of their successes ; as one inventor succeeds another, each one goes over again, to a great extent, the ground traversed by his predecessor, and discovers for himself the difficulties which overcame him. As one who has devoted many years of persistent effort to the development of the impulse-every-revolution engine, a short account will first be given by the author of the different engines built by him, with a statement of the difficulties peculiar to each.

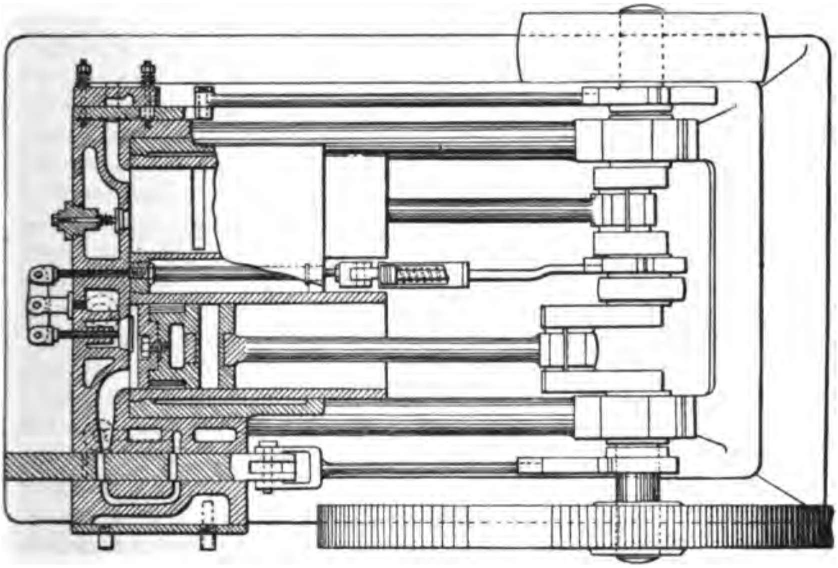


FIG. 118

The first engine built by the author is shown in fig. 118. It was designed by him early in 1878. There were two cylinders, each of 6 ins. diameter by 8 ins. stroke ; one being a pump cylinder, the other a motor cylinder. The pump crank followed the motor crank at right angles, and it pumped a mixture of gas and air into an intermediate reservoir through a check valve. From this reservoir the mixture was supplied to the motor cylinder by means of a slide valve. The mixture was compressed in the reservoir to a pressure of 70 lbs. per sq. in. above that of the atmosphere. The motor piston moved back to the end of the cylinder, leaving only a mechanical clearance of about ¾ in., and the exhaust valve was shut a little early in order to cushion the piston as is done in the steam engine. The motor piston moved forward from the end of its stroke about 2 ins., taking in the compressed mixture from the reservoir; the slide valve then cut off the supply, and ignition was effected by an incandescent platinum cage carried in the slide. This engine was exhibited at the Royal Agricultural Society's show at Kilburn in 1879, and was, the author believes, the first explosion compression gas engine ever built which gave an impulse for each revolution of the crankshaft. The author's object in the design was to utilise high compressions, and to obtain one impulse for every forward stroke of the motor piston. The engine at full power developed 4 BHP.

Two great difficulties were encountered after the minor difficulties requiring the invention of special igniting, governing, and starting gear were overcome: (1) back ignition into the compression reservoir; (2) shock in the motor cylinder. The difficulty of ignition spreading from the motor cylinder to the compression reservoir was serious, and was not completely overcome. The reservoir and the valves were made amply strong to stand the maximum pressure possible on explosion of the compressed contents; but as the reservoir contained about twelve charges from the pump, it was found that with full load on the engine it pulled up upon back ignition before the burned gases were cleared out of the reservoir. Back ignition occurred very seldom; but when it did it involved the stoppage of the engine. Various modifications were made to overcome this difficulty and that arising from shock, and the latter difficulty was obviated. The shock was due to the too rapid ignition of the contents of the motor cylinder, and was avoided by changing the shape of the compression space. This engine was capable of running at high speed, the igniter being very powerful. The incandescent platinum cage was so arranged that when once heated the platinum received heat enough from the consecutive explosions to maintain it in an incandescent state. As many as 300 [1] revolutions per minute have been attained with this engine with perfectly consecutive ignitions.

The next engine built by the author is shown at figs. 119, 120, and 121. In it the pump does not compress its contents to the pressure of the compression, but it simply forces the mixed gas and air into the cylinder while the motor piston is crossing the exhaust ports at the end of its stroke; the motor piston on its return stroke compresses the charge of gas and air into a clearance space at the end of the cylinder, when the compressed mixture is ignited, propelling the motor piston forward at every revolution of the crankshaft. The first engine of this type was designed by the author in 1880, and was built at the end of that year. The first engine exhibited was shown at the Paris Exhibition of 1881.

---

[1] A very high speed for 1878.

This Clerk engine has been extensively followed in Great Britain and on the Continent, and thousands of engines of this type have been put to work.

Description of Drawings.—Fig. 119 is a general view of the engine, fig. 120 a longitudinal section, and fig. 121 a sectional plan. In these drawings all the essential parts of the engine are represented; the sectional plan (fig. 121) shows the two cylinders, motor A and displacer B, in which work the pistons C and D, suitably connected to cranks not shown in the drawing, but on a common crankshaft. The motor crank is double; the displacer crank pin is fixed into an arm in the flywheel, and in the direction of motion of the engine is about a right angle or quarter circle in advance. The motor piston is shown at its extreme out-stroke, having passed over the exhaust ports E, E′, the piston thus serving as its own exhaust valve, and dispensing with any other, as shown; the displacer piston has moved half in and discharged a portion of the contents through the valve F (more distinctly seen in the other section, fig. 120) into the conical space G, which is so proportioned that the entering gases push before them the burned gases through the ports referred to, but without following them into these ports. By the continued movement all the gases in B pass into A and the space G; the capacities of the two cylinders are so related that as much as possible of the burned gases is discharged into the atmosphere,

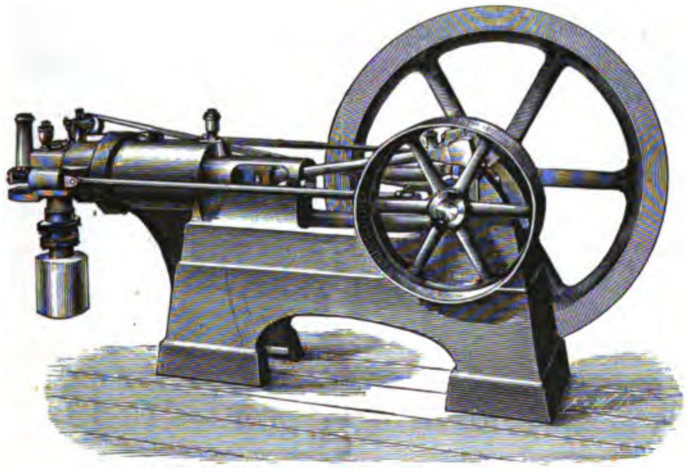


FIG. 119.—The Clerk Gas Engine

but without carrying away any of the fresh mixture containing unburned gases; this necessitates the mixture next the piston being somewhat more dilute than that next the inlet valve; but the commotion occasioned by the compression so far equalises this ununiform state of things that at half in-stroke the mixture in its weakened portions is quite capable of inflammation by a light or the electric spark. The piston D having completed its in-stroke, C has passed over the discharge ports and compresses the contents of the cylinder into the space G; when full in, and therefore completely compressed, the slide valve M has moved into such a position as to ignite the mixture; the maximum pressure is attained very rapidly before the piston can move appreciably on its out-stroke, and the piston is impelled forward under the pressure produced until it reaches the ports E, E′, when the contents are rapidly discharged, and the interior and exterior pressures equalised. Meantime the piston D being in advance of the motor has moved to the end of its stroke and is beginning to return; it has charged the cylinder B with a mixture of gas and air from the automatic valve H (fig 120), the communication being made by the pipe W (fig. 121). In the seat of this valve are bored a number of small holes passing into the annular space K K (fig. 120), which

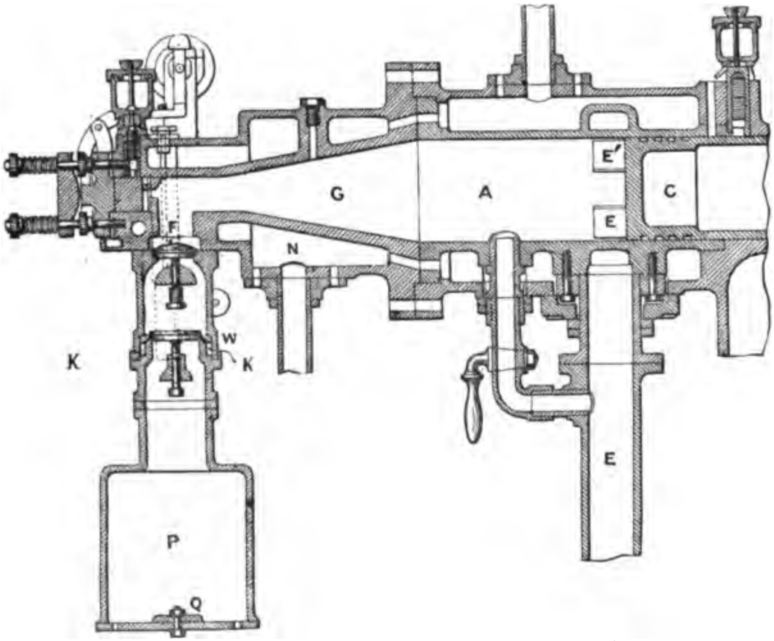


FIG. 120.—Longitudinal Section of Clerk Gas Engine

communicates with the gas cock L (fig. 121) through the passage shown, in which is situated the lift governing valve, not seen. Under the deficit

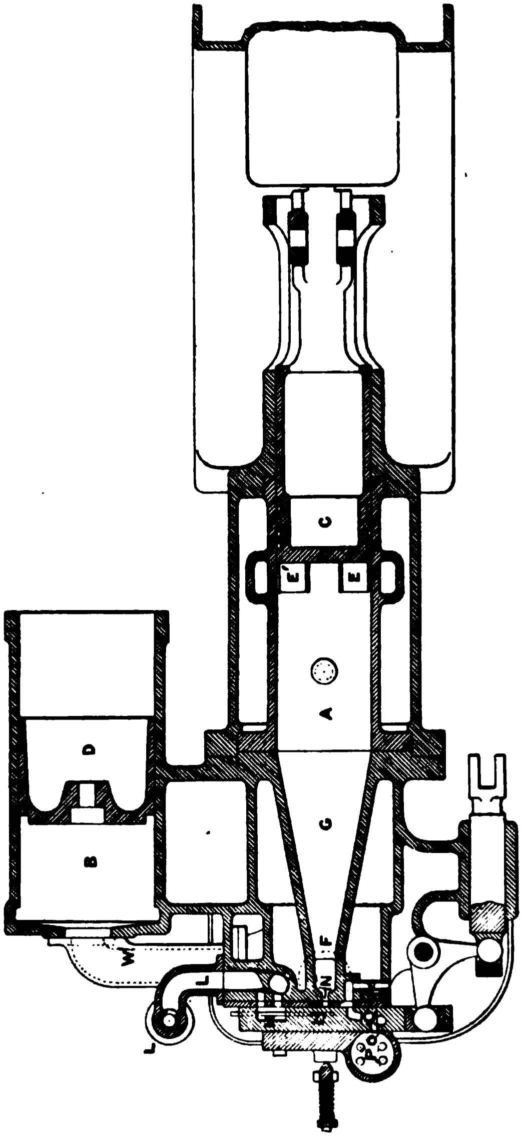


Fig. 121.—Sectional Plan of Clerk Gas Engine

of pressure caused by the movement of the displacing or charging piston, the valve is lifted and the air from the atmosphere rushes through; at the same time the gas passing through the holes mixes with it thoroughly, the proportion being determined by the relative areas of the holes and the space available for air by the lift of the valve.

The gases in B are under some slight compression before the complete discharge of A, but not sufficiently great to cause any material resistance; so soon as the pressure under the valve F is slightly in excess of that above it it lifts, and the gases then pass into G. The passage from the valve, which may be called the upper lift valve, is more clearly seen in fig. 120; the igniting hole is shown at N (fig. 121), and communicates at the proper time with flame in the cavity O, which has been ignited at the exterior flame P from a Bunsen burner.

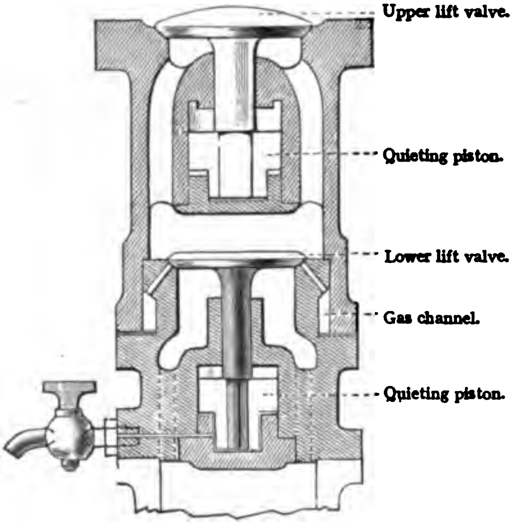


FIG. 122.—Section of Lift Valves, Clerk Engine

The two automatic valves charging the displacer cylinder and discharging into the motor cylinder are provided with quieting pistons, cushioning the blow on the valve seat and preventing rattle; they are similar to the dash pot contrivances used on Corliss' steam engines to check the snap of the steam valves, but, unlike them, are attached directly to the valve, instead of to the valve spindle and guide. The arrangement is very clearly seen at fig. 122; the lower valve has no spring, it returns to its seat by its own weight; but the upper valve requires to act more quickly and is pulled down by a spring.

The piston attached compresses the air before it, and the valve strikes its seat more rapidly, but without jar or recoil.

The igniting slide M is driven from an eccentric on the crankshaft through a bell crank and guide.

Diagrams and Gas Consumption.—The following tests give the results obtained from the Clerk engine in 1884; they are the usual trials which were made by Messrs. L. Sterne & Co. on all engines before leaving the works, and therefore represent fairly the economy obtained from these engines in ordinary work. They are from 2, 4, 6, 8, and 12 HP engines (nominal). The trials were made during 1884 at the Crown Ironworks, Glasgow, under the direction of Mr. G. H. Garrett.

TESTS OF THE CLERK ENGINES OF VARIOUS POWERS IN 1884. (*Garrett*)

—	2 HP.	4 HP.	6 HP.	8 HP.	12 HP.
Diameter of motor cylinder	5 ins.	6 ins.	7 ins.	8 ins.	9 ins.
Stroke	8 ins.	10 ins.	12 ins.	16 ins.	20 ins.
Diameter of displacer cylinder	6 ins.	7 ins.	7½ ins.	10 ins.	10 ins.
Stroke	9 ins.	11 ins.	12 ins.	13 ins.	20 ins.
Average revs. per min. during test	212	190	146	142	132
Average pressure (available) in motor cylinder in lbs. per sq. in.	43·2	63·9	53·2	60·3	64·8
Power indicated in motor cylinder	3·62	8·68	9·05	17·38	27·46
Power by dynamometer	2·70	5·63	7·23	13·69	23·21
Gas consumption in cub. ft. per IHP hour	29·28	24·19	24·23	20·94	20·39
Max. pressure of explosion in lbs. per sq. in. above atmos.	155	236	195	195	238
Pressure of compression in lbs. per sq. in. above atmos.	38	55	48	49	57
Displacer resistance	0·40	0·80	0·86	1·50	2·00
Gas consumed per hour by each engine at speed without load, cub. ft.	40	58	57	70	90

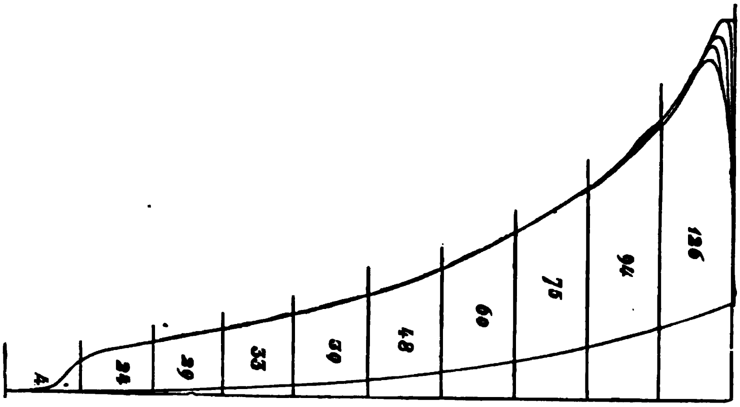
Figs. 123, 124, 125 are fair samples of the diagrams taken during the tests. Figs. 126 and 127 are diagrams from the displacers showing the displacer resistance.

Calculating from these diagrams the actual indicated efficiency, it comes to 16 per cent. of the total heat given to the engine.

The compression space in the Clerk engines was as nearly as possible one-half of the volume swept by the piston from the exhaust port to the end of its in-stroke. The theoretic efficiency according to the air standard is therefore

$$E = 1 - \left(\frac{v_c}{v_0}\right)^{\gamma - 1} = 1 - \left(\frac{1}{3}\right)^{0{\cdot}408} = 0{\cdot}36$$

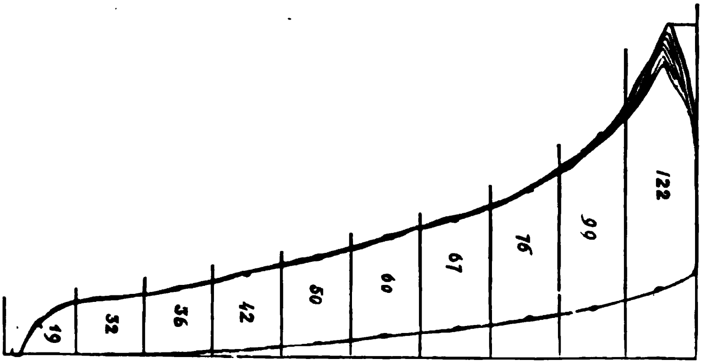
The compression is higher, and therefore the theoretic efficiency of this engine is higher than the Otto, but the difficulties of proportioning the two cylinders of the Clerk engine cause a small loss of unburned gas at the exhaust ports, so that the actual efficiency is similar to that of the Otto.



Nominal HP, 6; diam. of cylinder, 7″; length of stroke, 12″; No. of revs. 146; indicated HP, 9·05; consumpt. per IHP, 24·30 cub. ft.; consumpt. light, 57 cub. ft.; brake HP, 7·23; consumpt. per BHP, 30·42 cub. ft.; mean pressure, 53·2 lbs.; max. pressure, 195 lbs.; press. before ignition, 48 lbs.; scale of spring, $\frac{1}{100}$″ per lb.

FIG. 123.—Diagram from Clerk Gas Engine, 6 HP

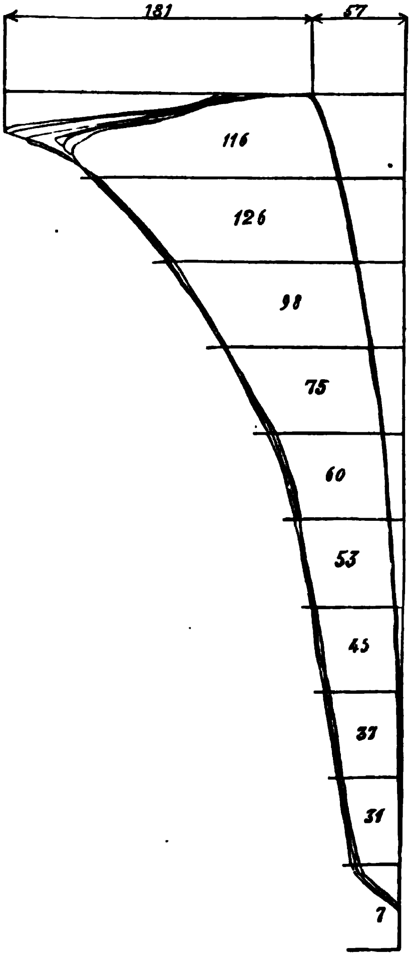
The mixture sent from the displacer cylinder into the motor and the space at the end of it contains 8 volumes of air with 1 volume of coal gas, but on passing through the upper lift valve and mixing to some



Nominal HP, 8; diam. of cylinder, 8″; length of stroke, 16″; No. of revs., 142; indicated HP, 17·38; consumpt. per IHP, 20·94 cub. ft.; consumpt. running light per hour, 70 cub. ft.; brake HP, 13·69; consumpt. per BHP, 26·58 cub. ft.; mean pressure, 60·3 lbs.; max. pressure, 195 lbs.; pressure before ignition, 49 lbs.; scale of spring, $\frac{1}{112}$″ per lb.

FIG. 124.—Diagram from Clerk Gas Engine, 8 HP

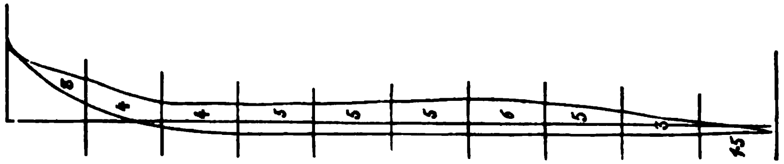
extent with the exhaust there contained, it is somewhat diluted; the heat acquired by contact with the products of combustion and with the sides of the cylinder expands the entering gases, and a temperature of not less than 100° C. is attained before the compression commences.



Nominal HP, 12; diam. of cylinder, 9″; length of stroke, 20″; No. of revs. 132; indicated HP, 27·46; consumpt. per IHP, 20·39 cub. ft.; consumpt. running light per hour, 90 cub. ft.; brake HP, 23·21; consumpt. per BHP, 24·12 cub. ft.; mean pressure, 64·8 lbs.; max. pressure, 238 lbs.; pressure before ignition, 57 lbs.; scale of spring, $\frac{1}{115}$″ per lb.

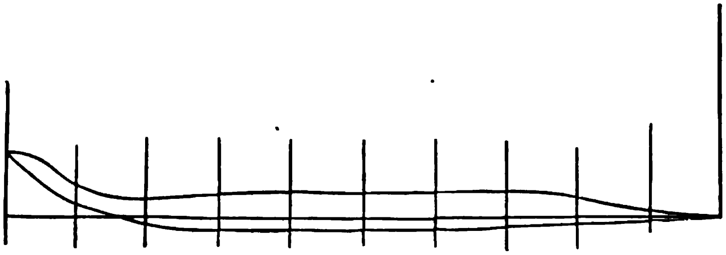
FIG. 125.—Diagram from Clerk Gas Engine, 12 HP

The result of this is that the displacer gases, being expanded, expel more of the exhaust gases through the discharge ports than would appear from the volume swept by the motor piston, from closing of the exhaust ports to complete in-stroke. If no expansion and no mixing occurred, the exhaust gases contained in the compression space would remain in front of the cooler explosive charge; but the heat increases the volume at least one-third, so that the volume occupied will be 1⅓ times the volume swept by either piston. The volume of cylinder plus space is 1⅓ vols. of cylinder, so that the actual exhaust gases present are ⅓ vol., or $\frac{1}{10}$ of the total gases present. But mixing must occur to a considerable extent and be made very complete on the return stroke during compression. The result of all this is the production of an explosive mixture which is explosive in every part of it, and of an average composition of one volume of coal gas in ten of the mixture. The proportion of burned gases present is very slight; the only reason why any should be left is the necessity of preventing any appreciable discharge of unburned gas at the exhaust ports. The mixture used is a comparatively rich one.



Scale of spring, $\frac{1}{32}$″ per lb.; mean pressure, 4·15 lbs.; IHP, 0·860; diam. displacer cylinder, 7″ stroke, 12″.

FIG. 126.—Diagram from Displacer Cylinder (Clerk Engine), 6 HP



Scale, $\frac{1}{32}$″ = 1 lb.

FIG. 127.—Diagram from Displacer Cylinder (Clerk Engine), 8 HP

In 1886 the author built another engine at Birmingham of the same type as the 1878 engine, but differing from it in having no reservoir between the pump and the motor. This engine is shown in elevation and plan in fig. 128, an end view in part section, showing the arrangement of the levers, and a sectional plan being given in fig. 129.

A motor diagram and a pump diagram are given in figs. 130 and 131.

Fig. 132 shows the relative positions of the motor and pump cranks at different positions at the moment of firing. This engine gave very economical results, and its running was satisfactory when the adjustments of the valves were accurately maintained. The difficulty of back ignition into the pump was overcome by diminishing the passage between motor and pump to the smallest possible volume.

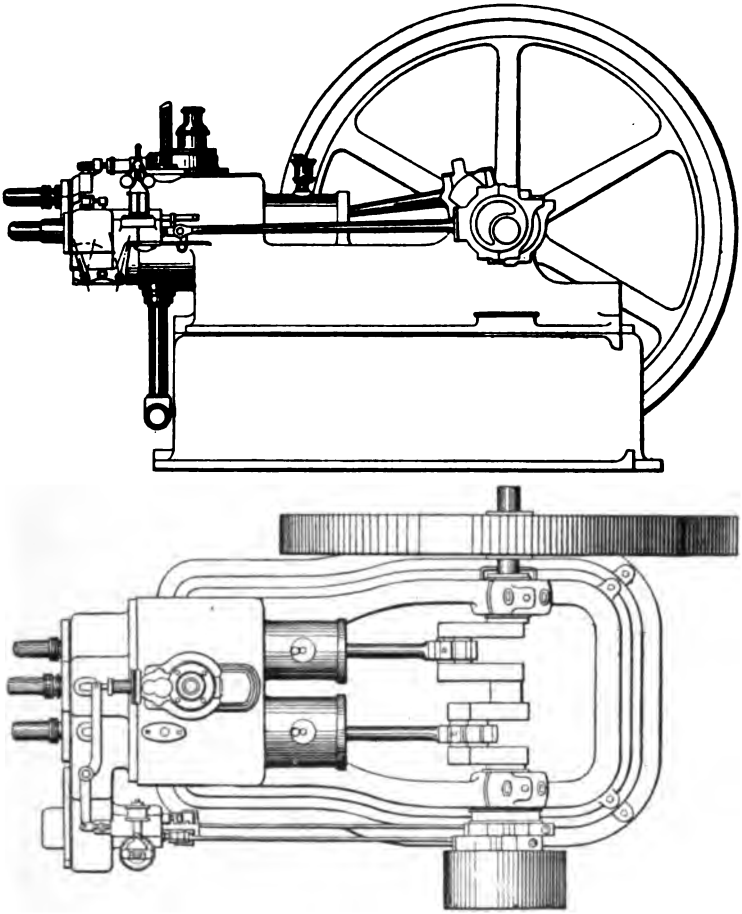


FIG. 128

When back ignition did occur the engine was not materially affected, as the pump at once took in a new charge of gas and air, and after compressing it, delivered it into the motor cylinder as the piston moved on its forward stroke. The motor piston moved on to the end of the stroke, leaving only mechanical clearance, and so discharged the whole of the exhaust products.

The governing was accomplished by opening a special valve from the pump cylinder to a reservoir of considerable capacity, closing the valve communicating between the pump and motor cylinders, and holding open the exhaust valve during the exhausting and charging strokes. The effect of these operations was to prevent the transfer of a charge from pump to motor; the pump piston acting to compress and expand the charge to and from the governing reservoir, while the motor piston alternately discharged the exhaust gases into the

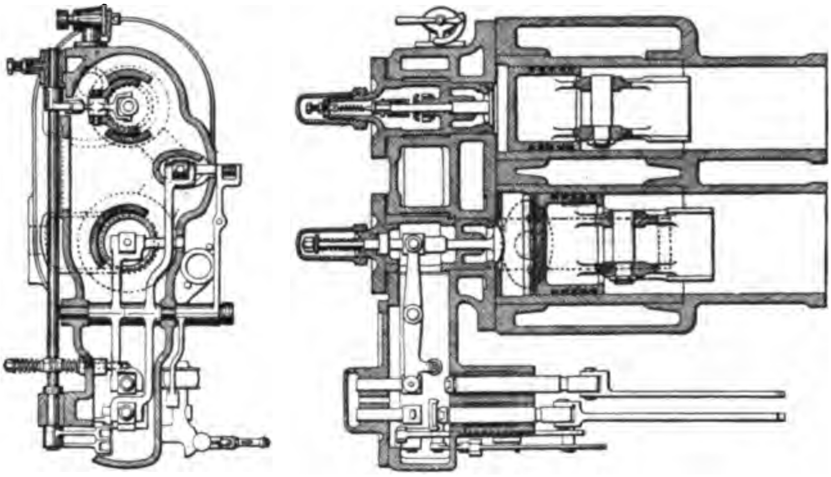
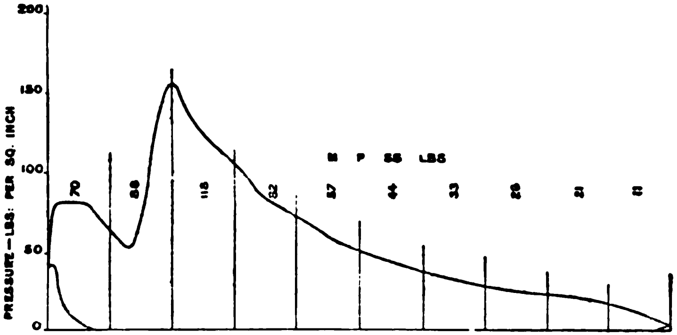


FIG. 129



Cylinder diam., 9"; stroke, 15"; revs. per min., 140; IHP, 10·71 at 54·9 lbs. per sq. in. mean pressure (average of several cards); gas consumpt. per IHP per hour, 20·1 cub. ft.; BHP, 8·96; gas consumpt. per BHP per hour, 24·1 cub. ft. (Birmingham gas).

FIG. 130.—Motor Diagram, Clerk 1887 Engine

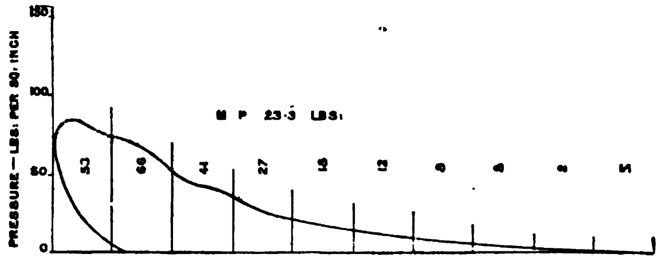
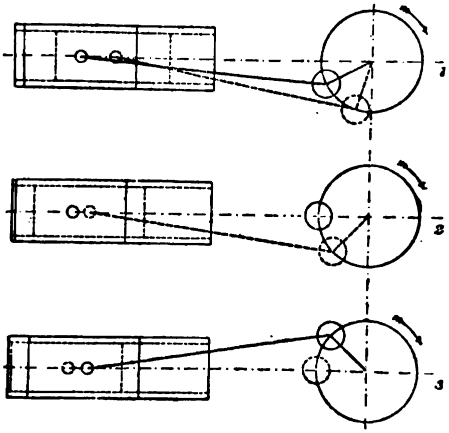


FIG. 131



The motor-crank and connecting-rod are shown in full lines and the pump-crank and rod in dotted lines; 1 is the position when the valve opens between motor and pump; 2 when the motor-piston is full in; and 3 shows the pump-piston full in and the motor-piston in the igniting position.

FIG. 132.—Diagram showing Positions of Motor and Pump, Clerk 1887 Engine

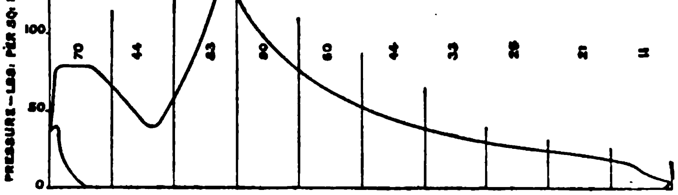
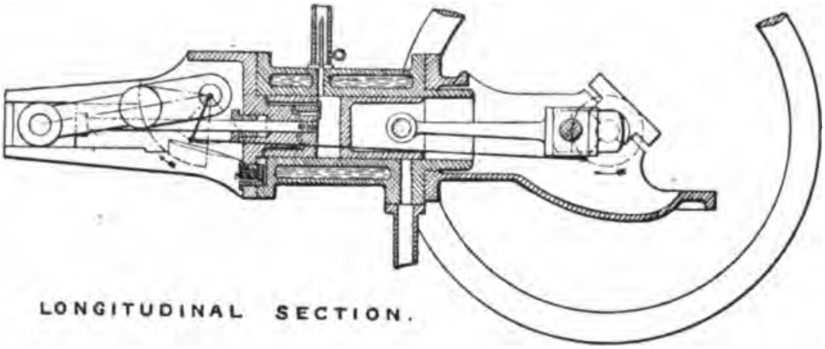


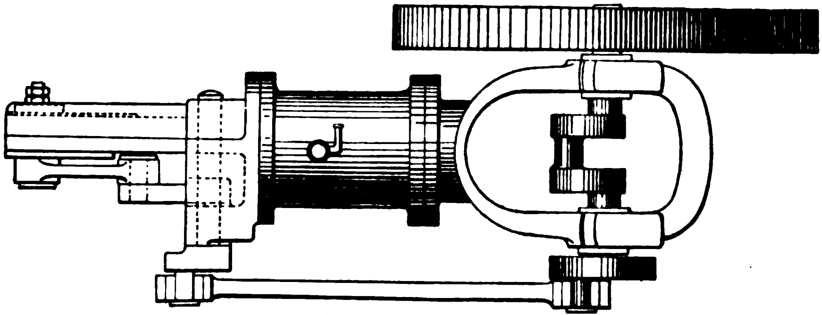
FIG. 133

exhaust pipe, and took them back to the motor cylinder. The governing was economical.

The engine is too complex to compete with advantage with small engines, but it may yet be adapted for large powers. It admits of any required compression and any required expansion of the gases after ignition.



LONGITUDINAL SECTION.



PLAN.

CLERK ENGINE 1889.

FIG. 134

To secure the best results, it is necessary that the ignition should follow the closing of the inlet valve from the pump to the motor with extreme rapidity, as a short delay in firing causes considerable loss of power. This is shown by fig. 133, where the ignition has occurred too late.

A considerable loss of power is thus caused by firing at a low instead of a high compression. To obtain the best results from this type of engine, the stroke should be long in relation to the diameter of the cylinder. With a large engine with a cylinder of, say, 24 ins. diameter, a stroke of 4 ft. would be desirable ; such an engine could readily be designed to give a consumption equivalent to 11 cub. ft. of Birmingham gas per IHP hour. Eighty revolutions per minute would be a suitable speed for it. The engine would be arranged to give an impulse every revolution when light as when loaded. This type of engine has also been arranged by the author to have a considerable clearance space at the end of the cylinder, which space should be swept out by pure air. With such a space, the difficulties of the engine are considerably diminished.

In 1889 the author designed the engine illustrated in figs. 134 and 135, and a small example of the type was constructed at Birmingham in 1890 for experimental purposes. Two pistons are employed in one cylinder ; one, the trunk piston, is the motor piston, and the second is the charging and exhaust expelling piston. The second is operated by a piston rod passing through a cover at the back of the engine cylinder ; this rod is attached to a cross-head working in guides, operated by a separate crank (1), connecting-rod (2), lever (3), rocking-shaft (4), and toggle-links (5) and (6). The motion is so devised that when the motor piston is out and crossing the exhaust port, the charging piston moves out and discharges the products of combustion from the cylinder by the front exhaust port. At the same time it takes into the cylinder a charge of gas and air on the other side. The two pistons then move back together and the charge is compressed ; after a time the charge passes from the back of the charging-piston to the front, and so occupies the space between the two pistons, and is ignited when the motor piston is full in. The motor-piston then moves out under the pressure of explosion, while the back piston remains practically stationary. This pause is brought

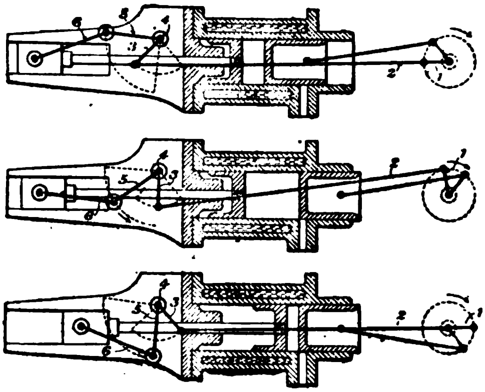


Fig. 135

about by the toggle-links crossing their centre, and it is sufficiently prolonged to enable the motor piston to practically complete its stroke before the back piston begins to move.

In the first position, fig. 135, the engine is as when the mixture is ignited. In the second the motor piston is just over-running the exhaust port and the charging piston about to move out. In the third the piston has returned, closing the exhaust port, and the charging piston is fully out, having taken in its charge at the back of the piston and discharged the exhaust at the front; the first position is then repeated, when the charge is compressed and ready to ignite.

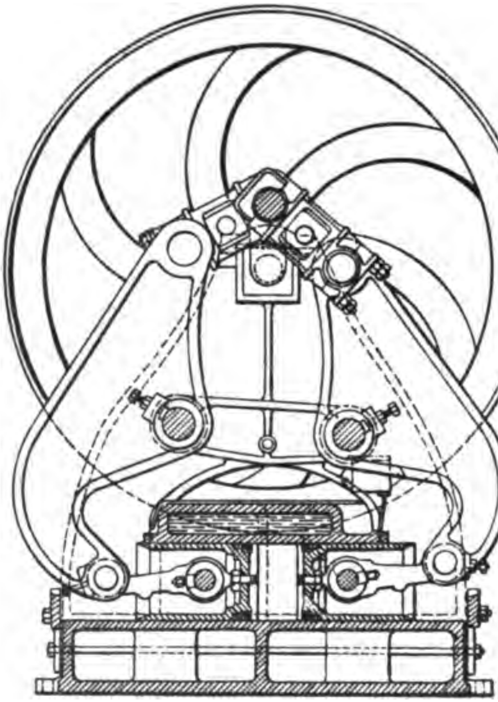


Fig. 136

The author has now given a short account of the leading engines built by him in his endeavours to solve the impulse every revolution problem, and he will proceed to shortly describe the work of Mr. James Atkinson, who has made a very determined effort to conquer the difficulties attending this type of engine. Mr. Atkinson's first gas-engine patent is dated 1880, but his first compression gas engine was exhibited in 1884. It was designed much on the lines of the author's

1878 engine, and the difficulties encountered by both were similar. Mr. Atkinson's first original solution of the difficulty of obtaining an impulse at every revolution in an explosion compression gas engine was made in 1885, a vertical section of the engine being shown in fig. 136. Fig. 137 shows the various positions of the two pistons which perform the operations of the cycle.

Two pistons *a* and *b* are connected with a common crank-pin, *c*, by levers, *d e*, and rods, d^1 e^1; short links, *x* and *g*, connect the lower ends of the levers with the pistons. In the first position the exhaust is completed. In the second the piston *a* has moved away from the other one to take in the charge of gas and air. The piston *b*, in the next position, has followed *a* and compressed the charge, while in the last the piston *b* has moved out and expanded the gases after explosion. The engine is very ingenious, and performs all the operations of charging, compressing, exploding, expanding, and exhausting in one revolution of the crankshaft. A consumption of 25 cub. ft. of London gas per brake horse-power hour at full power was claimed for this engine by its designer. It was found, however, that the linkage was troublesome mechanically, and that the consumption of gas when running without load was high. The engine was capable of expanding the hot gases down to a low pressure, but a great disadvantage was found in the shallow compression space in relation to the diameter of the cylinder. A shallow compression space approximates too closely to an ideal condenser or cooler, and so an engine with such a space may lose more heat than is compensated for by the extra expansion.

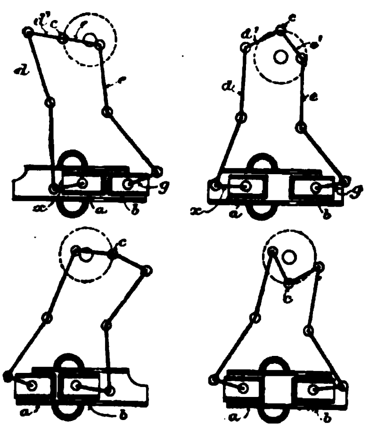


FIG. 137

Mr. Atkinson replaced this engine, which he called the 'differential gas engine,' by a very ingenious engine to which the name of the 'Cycle' engine was given.

The engine is shown in longitudinal section at fig. 138, in plan at fig. 139, and at fig. 140, at 1, 2, 3, and 4, are given the four principal positions of the linkage and piston, carrying into effect the operations of the engine. The piston makes two out- and two in-strokes for every explosion given, and in this feature the engine resembles the Otto, but there the resemblance ends. The piston is so coupled to the crankshaft that the whole four single strokes are performed during one revolution ; and moreover, the four strokes differ in length and range in the cylinder, so that while on one in-stroke the piston proceeds almost entirely to the end of the cylinder to sweep out practically the whole of the products of combustion, on the next in-stroke it stops

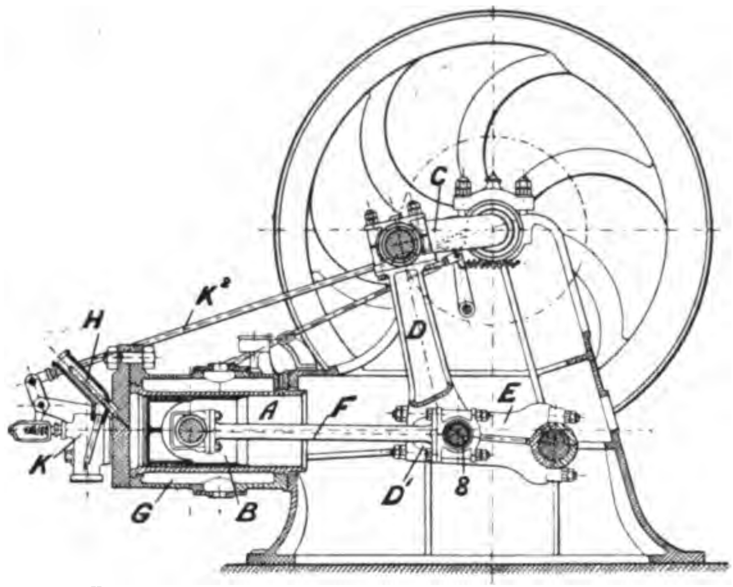


FIG. 138.—Atkinson Cycle Engine (longitudinal section)

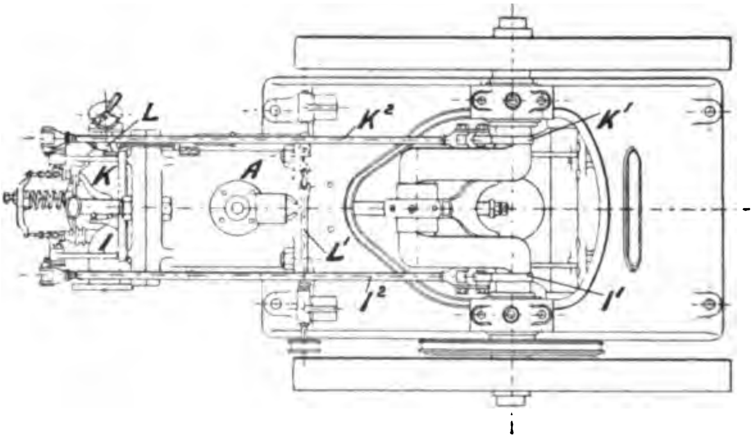


FIG. 139.—Atkinson Cycle Engine (plan)

short and leaves a considerable compression space; on one out-stroke also a short distance is traversed, and on the other out-stroke a longer stroke is made to obtain greater expansion. That is, during the exhausting in-stroke the piston moves close up to the cylinder cover; during the compressing in-stroke it leaves a considerable space; during the expanding out-stroke after explosion the piston makes its longest sweep; and during the charging out-stroke it makes a shorter sweep.

By these variations in length of stroke and position of sweep in the cylinder the piston not only sweeps out the whole of the products of combustion, but it also expands the burned gases beyond the volume existing before compression. The linkage invented by Mr. Atkinson to perform these operations is extremely simple and ingenious, and will be best followed by an examination of the diagrammatic illustrations 1, 2, 3, and 4 of fig. 140.

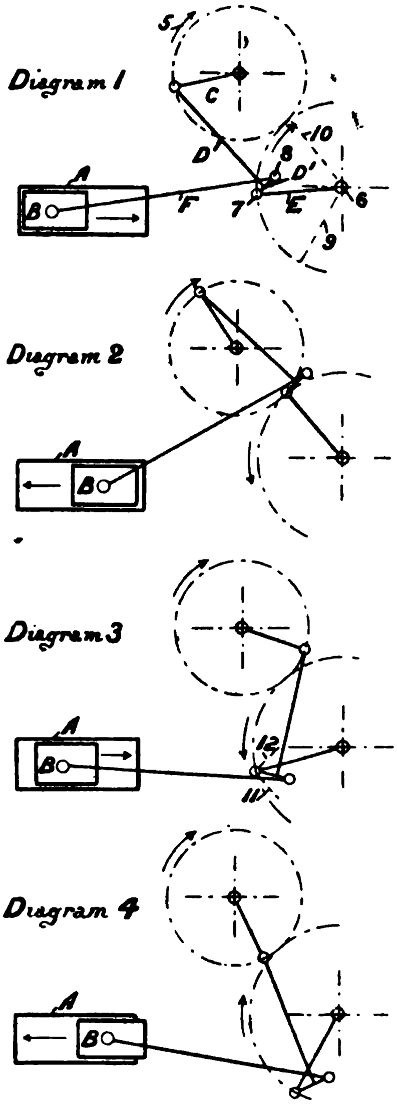


FIG. 140.—Atkinson cycle engine (four positions of linkage)

The cylinder A contains the piston B, which piston is connected to the crank C, which rotates in the direction of the arrow 5; the connecting-rod D carries a short lever D^1 rigidly attached to it and carrying a pin or centre 8, and to this pin or centre 8 is connected the second connecting-rod or toggle link F. By the rotation of the crank C the toggle lever E is constrained to oscillate on its pivoting point or centre 6, between the limits shown by the dotted lines 9 and 10. The centre 7 of the connecting-rod D and lever E thus describes the arc shown by the dotted line between the lines 9 and 10, and if the rod F were connected to the centre 7 the piston B would make two out- and two in-strokes for every revolution of the crank C, and the two strokes would be of equal or unequal length depending on the equal or unequal oscillation of the toggle lever E about a central position with regard to the connecting-rod F; but in this case the in-stroke of the piston B would always terminate at the same point, and so one stroke could not be arranged to clear out the exhaust gases, while another left the compression space. To produce this desired variation, Mr. Atkinson provides the short lever D^1, which oscillates about the centre 7, describing an arc between the dotted lines 11 and 12 about the centre line of the lever E. The position of the centre 8 relative to the centre line E depends on the position of the crank C in the crank circle, and the angle between the lines 11 and 12 depends upon the relative length of the connecting-rod D, as compared with the diameter of the circle described by the crank C, and also the angle between the lines 9 and 10.

In diagram 1 (fig. 140) the piston B is at its extreme in position, and all products of combustion have been expelled; the crank C rotates in the direction of the arrow 5; and in diagram 2 the piston B has made its out charging stroke, taking into the cylinder a charge of gas and air at atmospheric pressure. It is to be observed that in diagram 2 the piston B, although at the end of its charging stroke, still remains within the cylinder A; in diagram 3 the crank C has still further rotated, and now the piston B has attained the extreme in end of its compression stroke, the mixed gases are fully compressed and ready for explosion; the explosion takes place at the position shown in diagram 3, and, the crank C continuing to rotate, the parts at the extreme outward position of the piston B after expanding the gases assume the position of diagram 4. In this latter it will be observed that the piston B has travelled somewhat out of the cylinder A—that it has made a longer stroke than the compression stroke. The stroke made in passing from the position of diagram 4 to that of diagram 1 is the longest of all strokes, as in it the piston passes from the extreme out-position of expansion right into the cylinder cover, and sweeps out all the products of combustion. The next out-stroke is shorter, taking in the charge; the following in-stroke is shorter still, compressing the charge and leaving a compression space; then follows the longest out-stroke, that of expanding the gases after explosion. A comparison of diagrams 1 and 3 shows the reason of the difference of position of the piston B; although in both cases the toggle lever E is in practically the same position, in 1 the crank C is on one side of the crank circle, while in 2 it is on the other side, so that the lever D is thrown from the top of the centre line of the toggle lever E to a position under it, but the effect of the movement is to draw the piston forward from the cylinder cover. By studying the positions of the lever D and the positions of the toggle lever E from the diagrams, the action will be readily followed.

In the engine rated at 6 HP nominal the cylinder is 9·5 ins. diameter, and the four successive strokes are as follows:

1st (out-stroke) suction of gas and air charge	6·33 ins
2nd (in-stroke) compression of charge	5·03 ins.
3rd (out-stroke) working expansion after explosion	11·13 ins.
4th (in-stroke) discharging exhaust	12·43 ins.

The construction of the 6 HP engine is shown at figs. 138 and 139 in longitudinal section and elevation; A is the cylinder; B the piston; C the crank; D the connecting-rod to the toggle lever; E the toggle lever; D^1 the short connecting-rod lever; F the connecting-rod between the piston and the pin 8 on the lever D^1; G is the water jacket surrounding the cylinder and fitted with the usual openings for pipe connections to the tank; H is an incandescent igniting tube, open to the cylinder, and arranged to operate without timing valve in a manner to be described later on; I is the exhaust valve; and K the gas and air inlet valve (shown in plan, fig. 139); L is the gas valve. All three valves are of the usual conical-seated lift type, held on their seats by springs, and they are operated from the crankshaft by cams I^1 K^1 and rods I^2 K^2, in the usual way. The governor is indicated at L^1, and is of the rotating centrifugal type; it acts on a rod connecting between the actuating cam and the gas valve stem to cause the end of the rod to be withdrawn, and the gas valve stem missed, so leaving the gas valve closed for a stroke or a number of strokes. This also is a common device.

Diagrams and Gas Consumption.—A test of the first 'Cycle' engine constructed was made in April 1887 by Prof. W. C. Unwin, F.R.S., for the British Gas Engine Co., Ltd., the makers of the engine in London.

The engine was rated at 4 HP nominal, the diameter of the cylinder 7·5 ins., and the expansion or working stroke 9·25 ins.

The leading results obtained were as follows:

Indicated horse-power	5·563
Brake " "	4·889
Gas consumed in one hour	100 cub. ft.
Gas consumption per IHP hour	19·78 cub. ft.
Gas consumption per brake HP hour	22·50 cub. ft.
Efficiency of mechanism	87·9 per cent.
Heating value of gas in degrees C. Th.U. per cub. ft.	349·3

Professor Unwin accounts for every 100 heat units used by the engine as follows:

	Per cent.
Accounted for indicator diagram	20·62
Given to jacket water	19·37
Difference, exhaust gases, radiation, &c.	60·01
	100·00

An indicator diagram taken during the test is given at fig. 141, and in dotted lines on the same diagram is one taken by Dr. Slaby from a 4 HP Otto engine. This latter diagram was taken by Dr. Slaby during a test referred to at p. 6 of this work.

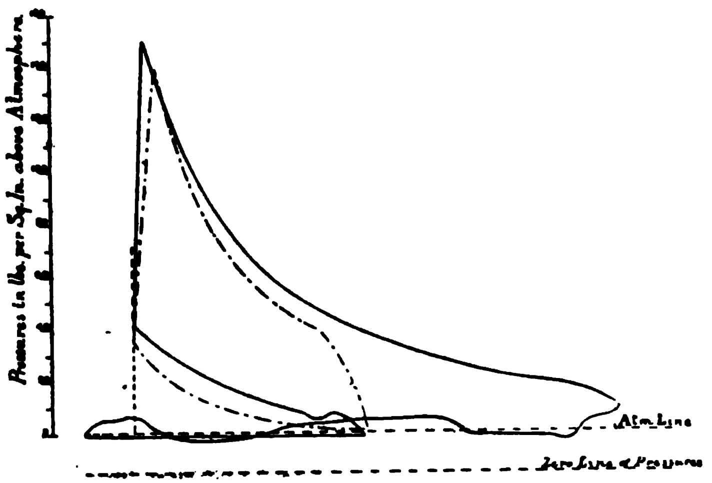


FIG. 141.—Atkinson Cycle Engine (Prof. Unwin's diagram)

The ratio of the expansion in the Otto engine was 2·7, as compared with 3·75 in Atkinson's; that is, in the Otto engine, the volume of the compression space being taken as 1, then the total volume behind the piston, when the piston was full out, was 2·7 volumes; the sweep of the piston was therefore 1·7 times the volume of the compression space; in the Atkinson engine, the volume of the compression space being 1, the volume swept by the piston during expansion was 2·75; the gases contained in, the compression space were thus expanded from 1 volume to 3·75 volumes. In the

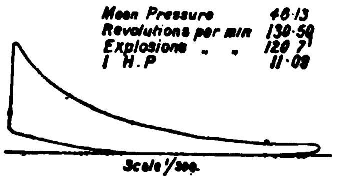


FIG. 142.—Atkinson Cycle Engine (Society of Arts diagram)

author's opinion, Professor Unwin's diagram, fig. 141, is hardly fair to the Otto engine, as it appears to somewhat exaggerate the amount of expansion obtained in the 'Cycle' engine compared with the Otto, and the diagram should be so corrected as to allow for the differing combustion spaces. The expansion, however, in the Atkinson engine is doubtless much greater than in the Otto, and accordingly the gases fall to a pressure of about 15 lbs. per sq. in. before the exhaust valve is opened.

An important series of tests was made by judges appointed by the Society of Arts, the late Dr. John Hopkinson, F.R.S., Professor A. B. W. Kennedy, F.R.S., and the late Mr. Beauchamp Tower, at South Kensington in 1888, of the Crossley, Griffin, and 'Cycle' gas engines, from which it appeared that the Atkinson 'Cycle' gave distinctly the lowest consumption. The principal results obtained were as follows:

SOCIETY OF ARTS TRIAL—ATKINSON ENGINE

Indicated horse-power	11·15
Brake ,, ,,	9·48
Gas consumed in cylinder in one hour	209·8 cub. ft.
Gas consumed for ignition in one hour	4·5 cub. ft.
Gas consumption per IHP per hour total	19·22 cub. ft.
Gas consumption per brake HP per hour total	22·61 cub. ft.
Efficiency of mechanism	85 per cent.
Heating value of gas in degrees C. Th.U. per cub. ft.	351·6
Revolutions per minute	131·1
Explosions per minute	121·6
Mean initial pressure above atmospheric	166 lbs. per sq. in.
Mean effective pressure	46·07 lbs. per sq. in.
Cooling water per hour	680 lbs.
Rise of temperature cooling water	50° F

The engine was rated at 6 HP nominal; the cylinder was 9·5 inches diameter; the suction stroke 6·33 ins.; compression stroke 5·03 ins.; working or expansion stroke 11·13 ins.; and exhaust stroke 12·43 ins.

The test giving these figures was of 6 hours' duration, and the engine was continually loaded to full power; indicator diagrams were taken every 15 minutes, and diagrams were also taken with light springs to find the power absorbed in the pumping and exhausting strokes. Fig. 142 is a diagram taken during this trial, and the leading particulars are marked upon it.

Fig. 143 shows an ideal diagram superposed upon an actual diagram; the ideal diagram is the one assumed by the judges in the Society of Arts trial as fairly corresponding with the actual conditions, lines have been straightened out, and curves made to follow a different law in order to obtain approximately correct figures for temperatures and heat volumes. Standard points A, B, C, D, E, F, and G have been taken, and the volumes existing behind the piston accurately measured at the various points. The point A, for example, represents the farthest in point when the piston is full back, discharging the products of combustion. The piston moves out from A to B, taking in the charge of gas and air; the piston then returns from B to C, compressing the charge to the pressure and volume indicated. The explosion then occurs, and during the rise of pressure and temperature the piston is supposed to be stationary, that is, the volume behind the piston is the same when the pressure attains its maximum as indicated at D as at the same point C, so that the heat added by the explosion is added when the gases are at constant volume; from D to E the piston is supposed to move out while the pressure remains constant, that is, the heat added from D to E is added at constant pressure. The hot gases are supposed to expand from E to F, following truly a definite curve in which pv^n = constant. In the diagram n is taken as 1·264, and the curve E F has the equation $pv^{1·264}$ = constant.

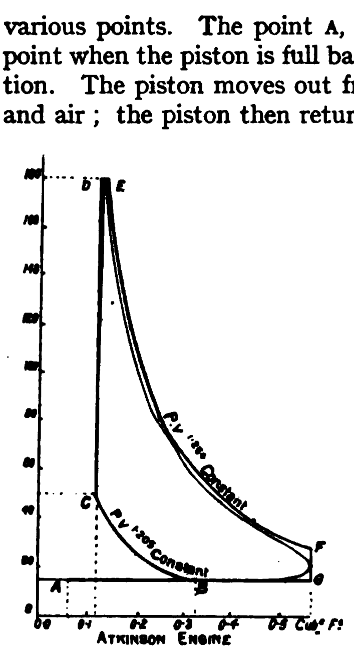


Fig. 143.—Atkinson Cycle Engine (Society of Arts actual and ideal diagram)

The value of n for the compression curve is 1·205, and assuming the specific heat of the charge the same as that of air, which assumption is very nearly true, then the adiabatic value of n should be 1·408; the curve of compression in this diagram is therefore below the adiabatic, and the charge is losing heat to the sides of the cylinder during compression.

The value of n for the expansion line E F is 1·264, which proves the curve to be much flatter than would have been given by the adiabatic expansion of a volume of air heated to the maximum temperature at E. The ratio of the specific heats of the expanding charge at constant pressure and constant volume has been calculated by the judges as 1·376, on the assumption of the presence in the charge of the products of complete combustion.

The table on p. 202 gives the pressure, volumes, and temperature at the various points of the ideal diagram, fig. 143.

The report gives full calculations of heating value of the gas used, specific heat of products of combustion, and many other details.

It is desirable to note, that the ratio of air to gas entering the cylinder is calculated as 1 volume gas to 9·33 volumes of air. It is unfortunate that the ratio was not determined by independent measurement of both gas and air by separate meters, as was done by Professor R. H. Thurston's American test, described on p. 8 of this work. Comparing the proportions of coal gas and air plus other gases present, it is interesting to note that in Thurston's experiments the entering charge contained 1 volume of gas to 7 volumes of air, but when mixed with the products of combustion in the compression space the average composition was 1 volume of coal gas to 9·1 volumes of other gases. The composition of the mixture in the two cases thus appears to be practically the same. In the Otto engine, however, the temperature of the charge was much higher before compression than in the Atkinson engine, as was also the temperature of compression; the maximum temperature of the explosion would appear to be higher in consequence.

—	Pressure, lbs. per sq. in. absolute	Volume in cub. ft.	Temperature, degrees C.
A	14·87	0·064	—
B	14·87	0·324	46·6
C	50·30	0·118	126·4
D	180·90	0·118	1182·5
E	180·90	0·135	1388·1
F	29·00	0·575	849·2
G	14·87	0·575	849·2

The report gives the heat account of the foregoing tests as follows:

	Per cent.
Heat turned into work as shown by indicator diagrams . . .	22·8
Heat rejected in water jacket	27·0
Heat rejected in exhaust, lost by imperfect combustion, and otherwise unaccounted for	50·2
	100·0

This gas consumption of 19·22 cub. ft. per IHP per hour, giving an efficiency of 22·8 per cent., was the best result so far as economy was concerned up to the date of the trial, September 1888.

General Remarks.—The Atkinson 'Cycle' engine was manufactured and sold by the British Gas Engine Company, London, from 1887 to the beginning of 1893, and during that period the author is informed that somewhat over 1000 gas engines were sold; the engine, however, notwithstanding the great ingenuity of its construction and its unrivalled economy of gas consumption, never became really popular. Difficulties were experienced with the linkage, which had at least five working pins as compared with the two pins of the ordinary connecting-rod, and these difficulties ultimately led the inventor to return to an engine of less uncommon construction, having only the ordinary crank and connecting-rod.

Mr. Atkinson, however, in his 'Cycle' engine proved the possibility of obtaining great economy in gas consumption by expanding the gases after explosion to a volume much greater than existed before compression. By his ingenious linkage he caused one piston to perform four strokes within one revolution of the crankshaft; he also proved conclusively a point for which the present author has long contended, namely, that better results are to be obtained in a gas engine by expelling the whole of the products of combustion from the cylinder than by retaining them.

The 'Cycle' engine has a very high piston speed for a given number of revolutions of the crankshaft, each complete stroke of the piston being accomplished in about one-quarter revolution, as will be clearly seen by inspecting the diagram fig. 140. This high piston speed, although advantageous so far as gas consumption was concerned, must have been detrimental to the smooth and long-continued satisfactory working of the engine, as the movements of the piston rods and links were of a kind which could not be conveniently balanced.

The engine was made in sizes up to about 30 HP brake, and the author understands that one 100 HP engine was constructed, but he is informed that this size was never placed upon the market.

Atkinson's 'Utilité' Gas Engine.—This engine was invented by Mr. Atkinson with the object of retaining all the economy of the 'Cycle' gas engine, while returning to the ordinary mechanical arrangement of piston, crank, and connecting-rod, which has had the sanction of engineers and the public, inasmuch as it is practically the only construction adopted in steam engines.

The linkage of the 'Cycle' engine, although most admirable from an experimental point of view, was not such as an engineer would adopt in a high-speed or even a high-power engine, and although it served its purpose by proving to demonstration many interesting points, yet the present author was much pleased to see Mr. Atkinson depart from it.

The 'Utilité' engine never attained any real commercial importance, as the British Gas Engine Company gave up the business shortly after they had begun its manufacture. The Otto cycle had just then taken so firm a hold upon the public that it appeared useless to sue for popular favour with any impulse-every-revolution engine, however good.

The engine is of the greatest interest to engineers, however, as it proves how great economy can be obtained with an impulse-every-revolution engine.

The 'Utilité' engine resembles in many points the Clerk and Robson engines. One side of the piston operates as a pump and pumps air into a chamber at low pressure, from which it flows through a valve into the power side of the cylinder, and displaces the exhaust gases before it through a port or ports uncovered by the forward movement of the piston. The cylinder thus contains a quantity of air, which is compressed by the piston on its return stroke, and charged during compression with a gas and air mixture, the gas, however, in the mixture being present in proportion too great to be explosive.

The trunk piston operates in the cylinder connected to the crankshaft by the connecting-rod. The whole front of the cylinder and the rod and crankshaft are enclosed within a casing, and a back cover is arranged to contain the compression or explosion space. A chamber or casing connects by a pipe with an automatic inlet valve, and another automatic inlet valve admits air to the casing or chamber. A pump is operated by an eccentric on the crankshaft, and it takes in a charge of gas and air by way of a valve and the gas cock, and discharges the mixture at the proper time by way of the valve. This piston overruns the exhaust port at about half-forward stroke, and the port is controlled by a piston valve, so that, although the piston uncovers the exhaust port at mid-stroke, yet the exhaust gases are not discharged to the atmosphere till the exhaust valve is opened at the termination of the out-stroke.

The action of the engine is as follows: On the in-stroke the piston draws into the chamber a charge of pure air by way of the air valve, and on the out-stroke it compresses this charge to a pressure of about 5 lbs. per sq. in. When the piston is full forward the exhaust valve is opened, the pressure within the working cylinder falls to atmosphere and the pressure in the chamber lifts the charge valve, when air rushes by way of a pipe through it and enters the cylinder, clearing out the exhaust gases through the exhaust port and valve, which is then open. The piston returns, discharging the rest of the exhaust gases through the port until the piston crosses that port, when it begins to compress the air charge. Just as the port is closed, the gas and air pump begins to discharge its contents, a mixture of air and gas, into the combustion space of the cylinder by way of the gas and air valve. The gas and air mixture in the pump has too little air to make the charge explosive, and so it is impossible for the mixture to ignite in the pump. The gas being already mixed with air only requires the addition of a further quantity to become explosive, so that by the time the charge is compressed the gas is almost uniformly mixed with the air, and is in a state to produce a powerful explosion. The mixture is expanded by this arrangement to a volume after explosion and expansion much greater than the volume existing before compression, and so considerable advantage is obtained in economy.

The cycle of operation of this engine is very similar to that of previous engines, but Mr. Atkinson, by his thorough knowledge of gas-engine detail, has obtained results, he informed the author, which have surpassed those previously obtained with the 'Cycle' engine.

The Day Gas Engine.—This engine uses the same cycle of operations for charging the working cylinder as was adopted in the Tangye-Robson gas engine of 1880, and also in the Stockport engine of 1884, but the inventor ingeniously dispenses with all valves and valve gear such as cams or eccentrics, and he uses the crank case as a pumping chamber.

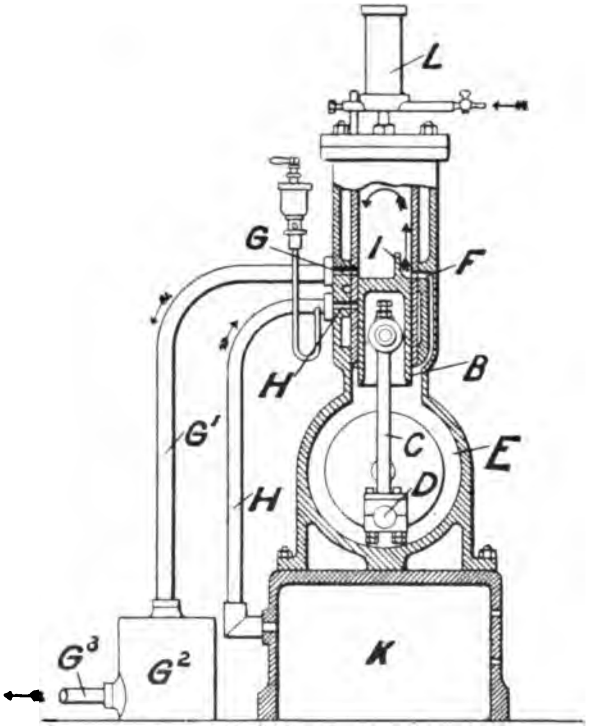


FIG. 144.—Day Gas Engine (vertical section)

Mr. Day's first patent is dated 1891.

The engine in one form may be described as valveless, and its only moving parts are piston, connecting-rod, and crankshaft; there is absolutely no valve used except a governor valve. Fig. 144 is a sectional elevation of one form of the engine, which is of the vertical inverted cylinder class, having the power cylinder overhead.

The piston B operates the crankshaft D by means of the connecting-rod C. The crankshaft operates in a closed chamber, E, which chamber serves as a reservoir for gas and air mixture. Three ports are arranged in the side of the cylinder, respectively F, G, H; F is the charge inlet port admitting the charge to the cylinder, G is the exhaust port allowing the discharge of the exhaust gases from the cylinder, and H is the air inlet port to permit of the admission of air from the external atmosphere. The charge inlet port F communicates with the crank chamber E, and opens to the cylinder opposed by the lip or projection I on the piston B. The exhaust port G connects by the pipe G^1 to the exhaust chamber G^2 of usual construction, and the chamber G^2 discharges to the atmosphere by the pipe G^3. The air inlet port H connects by pipe H^1 to the base of the engine K, so as to quieten the suction.

The action is as follows. On the up-stroke of the piston B the pressure of the gases in the chamber E is reduced to about 3 or 4 lbs. below atmosphere; at or near the end of the up-stroke the lower edge of the piston uncovers the air inlet port H, and air rushes into the chamber to bring the pressure up to atmosphere; gas is also admitted from the separate governor valve referred to, so that the chamber E becomes charged with a mixture of gas and air. On the down-stroke of the piston B the contents of the chamber E arc compressed to 3 or 4 lbs. above atmosphere, and at the termination of the down-stroke the port F is uncovered as shown in fig. 144. The exhaust port G has been crossed by the piston B somewhat earlier in the down-stroke, sufficiently early to allow the hot gases from the previous explosion to discharge to atmospheric pressure before the port F is opened. The charge then flows from the port F under slight pressure, strikes against the lip or baffle plate or projection I, and is deflected as shown by the arrows, so that it flows in a stream to the top, then turns and fills the cylinder, expelling the remaining exhaust gases by the port F. The piston B then returns on its up-stroke, and compresses the charge into a space at the top of the cylinder to a pressure of about 50 lbs. per sq. in. above atmosphere. The hot tube L then ignites the compressed charge, timing the explosion by the position of the incandescent part in a manner which will be explained more fully later on. The piston then makes its downward stroke under the pressure of the explosion. By these operations an impulse is obtained at every revolution, as in the Clerk, Robson, and Stockport engines.

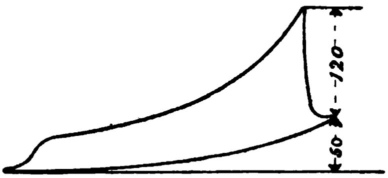


FIG. 145.—Day Gas Engine (diagram)

Loss of power is caused by the absence of an inlet suction valve to the space E, and in later engines a suction valve is provided.

This Day engine has the peculiarity that it can be run in either direction; this is possible because of the absence of timing valves or valve gear operated from the crankshaft.

Diagrams and Gas Consumption.—Fig. 145 is an indicator diagram from this type of engine rated at 1 HP nominal. The diagram shows an indicated power of 3·3 horse at 180 revolutions per minute; the cylinder is 4·5 ins. diameter and 7½ ins. stroke. The author has not obtained the gas consumption, but there seems no good reason why the results should be better than those obtained with Tangye's Robson gas engine. That engine gave for the small powers a gas consumption of 40 cub. ft. per BHP, with an average available pressure of about 45 lbs. per square inch. The diagram given shows an average pressure of about 45 lbs. per square inch, and is probably lower, as allowance should be made for the work of charging.

OTHER TWO-CYCLE ENGINES

Many English and Continental engineers and inventors have aided in the attempt to solve the two-cycle engine problem, but the foregoing description deals with all fundamental types; the names, however, of Robson-Tangye, Williams-Stockport, Campbell, Simon, and Beechey, among English inventors, and Koerting and Lieckfeld among Continental engineers, may be mentioned as illustrating the widespread attempt to overcome the difficulty of infrequent impulses.

At present the Clerk 1881 and Day 1891 types of two-cycle engine alone remain represented among modern engines.

The Clerk type has been largely followed by Messrs. Koerting of Hanover in large engines and to a small extent in certain small petrol engines such as the Dolphin (see Chap. XI).

The Day type is largely followed in petrol engines for launches, principally in America, but it is not used for large engines.

Large marine Diesel engines are now built of the two-cycle type, following the Clerk arrangement of separate charging cylinder.

Large gas engines are also built of the Oechelhauser design which follow a modified Clerk construction.

THE KOERTING TWO-CYCLE ENGINE (CLERK CYCLE)

Two-cycle engines built by Messrs. Koerting and their licensees compete strongly with large four-cycle engines, as over 200,000 HP have been supplied between them. It is proper then to describe the Koerting engine first, but before doing so it is desirable to state some points in which Clerk cycle engines (including Koerting and others) differ from the Otto engines.

In the Clerk cycle engines the considerations which govern power and economy very closely resemble those of the Otto cycle, but there are several points which require to be carefully considered (points, indeed, of considerable difficulty) with regard to which the Otto cycle is a far easier cycle than the Clerk. In the Clerk cycle the charging has to be accomplished in the motor cylinder, while the crank is passing through an angle of about 80°. Sometimes a little larger angle is allowed, but, roughly, 80° of the crank angle is the limit for charging of the motor cylinder. Because of this, much larger inlet valves and very much larger discharge areas are required in the Clerk than in the Otto cycle. In the Otto cycle the charging stroke occupies not only the whole of one stroke, which amounts to 180° of the crank movement, but in addition a further 40°, which permits the inlet valve to be held open considerably over the out centre, and also to be opened a little before the centre on the in-stroke. In consequence, the Otto type of engine allows three times the time interval to charge the cylinder, at a given rate of revolution, than it is possible to allow in the Clerk cycle. For a given valve area, the velocity of charge entrance in the Otto cycle is about a third of that in the Clerk cycle. This means that the Clerk cycle engines are more difficult to charge, and require greater power expenditure to charge their cylinders than the Otto cycle. This is one weakness of the Clerk cycle as constructed at present. Then there is this further point. In the Otto engine, when the piston moves out, taking in its charge, there is no question of any possible discharge at the exhaust ports, because the piston is sucking in the charge by a partial slight deficit of atmospheric pressure, and there is no exhaust port open through which fresh charge may be lost. In the Clerk cycle the proportions of displacer and motor cylinder have to be very accurately ascertained, otherwise part of the charge entering the cylinder is apt to pass right down through the centre of the exhaust gases which are being displaced, and pass out of the exhaust port. That was overcome in the early Clerk engines by a peculiar conical shape of cylinder end, which has been since consistently adhered to in all the engines operating on the Clerk cycle. That difficulty, however, is met in two ways. One way is to put in a smaller charge than in the Otto cycle, but that has the disadvantage of leaving too much exhaust gas, and also giving a smaller power of engine. Consequently, every designer of the Clerk cycle engine attempts to get in the full charge. The best method is to send into the cylinder, first a good heavy charge of air to displace the exhaust products, and then to follow it with a somewhat strong charge of gas and air. That is what the author did in 1881, and that is what is being done to-day in all the large gas engines. That, however, is a somewhat difficult thing to do. The consequence is that if one of these cylinders be charged as fully as it would be in Otto's cycle, some proportion of gas is lost at the exhaust ports, and although in a small engine with a comparatively light load the economy very closely approaches the best Otto economy, yet the maximum efficiencies that are possible with the Otto cycle have not been obtained with any small two-cycle engines.

In two-cycle engines it is evident that for a given revolution rate the cylinder and piston are exposed to the high temperatures of the gaseous combustion and expansion for double the time of the four-cycle. Against this, however, there is the fact that the hot exhaust gases are discharged from the cylinder at the time of closing the exhaust port about 40° to 45° of the crank on its return stroke, while in the four-cycle engine the hot exhaust is in contact with the cylinder and piston during the whole return stroke. The mean heat flow per second is, however, considerably higher in the two-cycle engine. Consequently, this engine requires even greater attention to cooling than the four-cycle. With greater heating comes greater liability to pre-ignition, but this and other troubles were fully realised by the author when he first designed the engine, as will be readily seen by means of the following extract from his patent specification: [1]

'In one modification of my improved motor there are two single-acting cylinders provided with pistons connected in the ordinary way to cranks on one shaft. In one cylinder a mixture of gas or vapour and air is ignited and power developed, but the other cylinder is employed to effect displacement only, and the two cylinders are hereinafter distinguished as the power and the displacement cylinder. The air and gas or vapour enter at one end of the power cylinder, and the exhaust takes place by ports so situated as to be passed by the piston when approaching the end of its stroke in its movement from the entering end. The capacity of the displacement cylinder is considerably larger than the product of the area and stroke of the power piston; and when the displacement piston is moving in one direction it draws in air and gas or vapour through a check valve, but the gas or vapour is cut off as soon as the quantity of mixed air and gas or vapour is about equal to the product of the area and stroke of the power piston; and only air is drawn in during the remainder of the stroke of the displacement piston. On the return stroke the displacement piston forces the contents of its cylinder through a check valve in the power cylinder, the unmixed air last drawn in first entering the power cylinder. The cranks of the two cylinders are so placed relatively to each other that whilst the contents of the displacement cylinder are being forced into the power cylinder the exhaust ports of that cylinder are still open, and not only do the burnt or used gases of the previous stroke pass out, but also the unmixed air which enters in advance of the fresh charge. This portion of unmixed air insures the cleaning out of the power cylinder and prevents any ignited matter from remaining in the cylinder to prematurely ignite the fresh charge. On the return stroke of the power piston, and after it has passed and closed the exhaust ports it compresses the mixture of air and gas or vapour into a clearance space at the end of the cylinder, and the compressed charge is ignited at about the commencement of the succeeding stroke. The ignition is effected by means of a slide worked by an eccentric on the crankshaft, the details of and connections with this slide being hereinafter particularly described. Besides having passages and ports which have to do with the power cylinder, the slide is made with a cavity or port operating in connection with ports and passages through which the combustible gas or vapour passes on its way from supply pipe to the valve through which it is drawn into the displacement cylinder, the slide causing the gas or vapour to be cut off at such period of the indrawing stroke as to render the later portion of air drawn in unmixed with gas or vapour. The gas or vapour supply passage is also fitted with a valve controlled by a speed governor.

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[1] English patent, Dugald Clerk, No. 1089 of 1881: 'Improvements in Motors worked by Combustible Gas and Vapour.'

'In other modifications of my improved motor two or more sets of the single-acting cylinders may be connected to one crankshaft, or the cylinders may be made double-acting by duplicating such parts as may be necessary.'

Messrs. Koerting have considered all these points and have successfully met all difficulties in engines of large dimensions. In the early Koerting two-cycle engines about 1901 the cylinder liner and water jacket casing were cast in one, as the makers then considered that the difference between the inner and outer expansion was not sufficient to unduly stress the metal of the casing.

Messrs. Mather & Platt in 1902 undertook the manufacture of the Koerting engine in England, and they at first built to the German design.

Fig. 146 is a sectional plan of a Koerting engine of German design as built by them in 1902 ; fig. 147 is a longitudinal section through the cylinder, and fig. 148 a transverse section through the combustion chamber at the inlet valve, showing the gas and air passages and the gas and air pumps.

This engine differs from the Clerk in that it is double-acting, so that the main crank receives two impulses per revolution, like the steam engine.

Instead, however, of having a single pump or displacer cylinder,

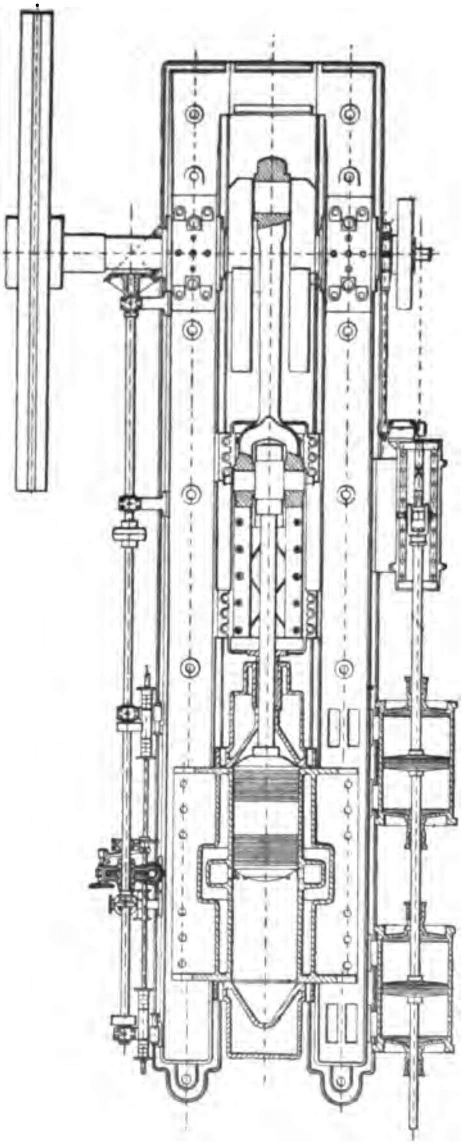


FIG. 146

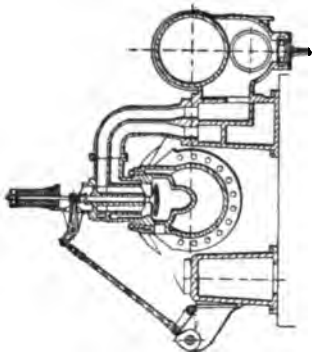


FIG. 148

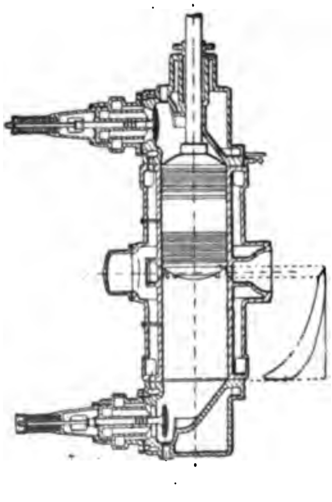
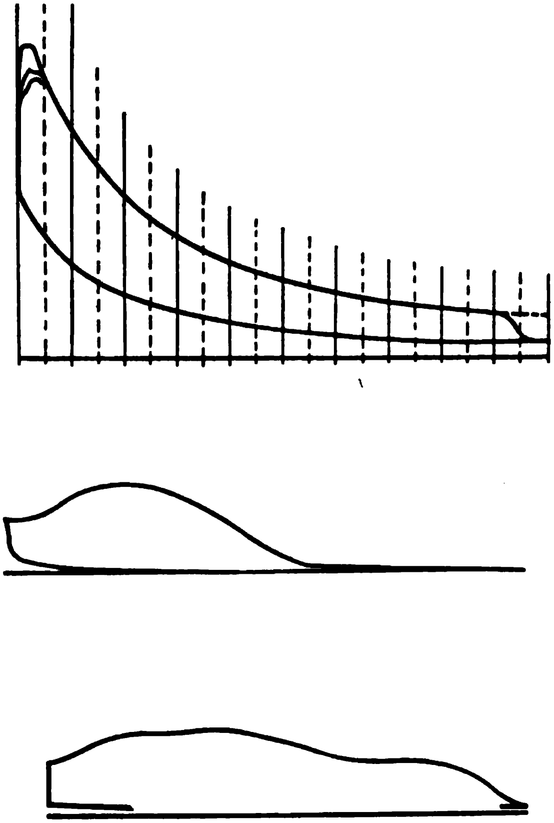


FIG. 147

two cylinders are provided, both double acting, one for gas and one for air.

In the illustrations the piston is shown at one end of its stroke, having overrun the exhaust ports and allowed the pressure to fall to atmosphere.



Power cylinder: dia. 24·8″, stroke 43·3″
Gas pump: dia. 25·6″, stroke 33·5″
Air pump: dia. 25·6″, stroke 33·5″
Revolutions: 100 per minute.
Engine designed to use blast furnace gas of 112 B.Th.U. per cub. ft. calorific value.
Mean effective pressure from power diagram = 86·3 lbs. per sq. in.

FIG. 149

Power and pump diagrams are shown at fig. 149 which were kindly given to the author by Mr. Ernest Koerting in 1902; from these it will be seen that the pressure begins to fall to atmosphere at the point where the piston overruns the ports, and it reaches atmosphere or very near it about 20° movement of the crank.

The pump or displacer crank is arranged about 100° in advance of the main crank like the Clerk engine, and so the pump pistons have travelled a short way on their discharging stroke, as is clearly seen on the air pump diagram by the rise of air pressure; in the gas pump diagram the gas pressure has not risen because the gas is by-passed to prevent a material change of pressure till later in its stroke. Thus a considerable volume of air first flows into the cylinder and displaces and cools the hot exhaust gases, so that later in the pump stroke, when gas flows from the pump to mix with the entering air, it impinges on the cool air within the cylinder and thus back ignition is avoided. It is very important in large gas engines to prevent the formation of inflammable mixture in chambers or passages; no mixture should be formed till the gases pass into the cylinder through the inlet valve. In a small engine the gas and air may be mixed in the pump or displacer, but in large engines the mixed charge should only be formed as it enters the cylinder. In a small engine a back ignition into the pump is not a serious matter, in a large engine it is.

When the charge has entered—air first and then gas and air—and has displaced the exhaust products, the main piston closes the exhaust ports by a crank movement of 40° to 45° from the centre. The compression then proceeds and ignition and expansion take place, so that an impulse is given at every stroke of the piston. The air pump valves are arranged to discharge the full air charge at every stroke whether the engine be light or loaded, but the gas discharged from the pump varies in amount determined by the governor, which allows gas to be passed back to the inlet side of the gas pump piston throughout more or less of the pump stroke. At full load the valve arrangements of the gas pump are so designed that for half stroke gas flows back to the inlet side, so gas pressure on the piston rises but little. At full load the latter half of the gas piston stroke forces gas through its separate passage into the entering stream of air and so forms mixture within the cylinder. As the load gets less the governor keeps the by-pass open so that the gas piston is later and later in forcing gas into the cylinder. By this device the mixture of gas and air admitted to the cylinder as it flows through the inlet valve is of constant proportion because that is determined by the relative areas of the two pistons, gas and air. The volume of air is constant, but less and less mixture is forced in. This mixture, however, fills the narrow end of the conical space, so that it is easily fired by the ignition near the inlet valve.

Fig. 150 shows in diagrammatic section the relative disposition of the separate gas and air passages, the gas and air pump valves and one method of by-passing. In the early designs piston valves were used and the gas was by-passed by an internal piston valve working within the main piston valve and under governor control. As this arrangement has been improved on by Messrs. Mather & Platt, it is unnecessary to describe the early devices in detail.

Messrs. Mather & Platt state that the following are the advantages claimed by them for this engine :

'1. Perfect scavenging with cool, fresh air, whereby pre-ignition by contact with products of combustion is entirely prevented, and lubrication much facilitated.

'2. Absence of heavy exhaust valves, which are usually a source of weakness, and liable to fail in their action owing to the intense heat to which they are subjected by the passage of exhaust gases.

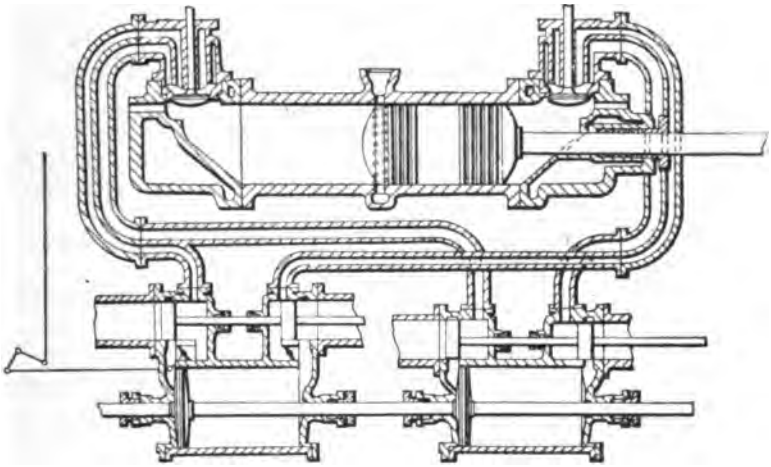


FIG. 150

'3. Removal from the power cylinder of up and down strains due to the resultant of impulse and resistance on the crosshead pin, which in this case is entirely outside the cylinder : for the same reason the crosshead pin is kept absolutely cool, and easily lubricated, giving a marked advantage over trunk piston engines.

'4. By obtaining an impulse each stroke, the diameter of the cylinder is just half of what is required for an "Otto" cycle engine of the same power ; the load on the working parts and the weight of the same are therefore small compared to the power given out ; also, since the contents of a cylinder vary as the square of its diameter, but the surface only varies directly with the diameter, it follows that for a given amount of jacket water the cooling results are more satisfactory with the small than with the large engine.

'5. Great steadiness of running, owing to the fact that just four times as many impulses are transmitted as in the case of a single-cylinder "Otto" cycle engine. Lighter flywheels can be used, and thus unnecessary weight and friction are avoided. As steady running and as high a degree of cyclic regularity are attained as with a steam engine.

'6. The engine can be started almost instantaneously; lubrication is forced and practically automatic throughout; the design generally lends itself to hard and continuous running, and when overhauling becomes necessary, this can be done with the greatest ease and in a very short time.'

It will be seen from the diagram, fig. 149, that in this particularly early engine the power cylinder was 24·8 ins. diameter and the stroke was 43·3 ins. The mean pressure on the power diagram was 86·3 lbs. per sq. in.

Mr. Junge[1] in his valuable book describes an interesting test of a 600 HP double-acting two-cycle Koerting engine made in October 1904, from which the author has obtained the following particulars:

TEST OF AN EARLY 600 HP KOERTING DOUBLE-ACTING TWO-CYCLE ENGINE.
(*Junge*)

Engine dimensions:
Power cylinder 29·7 ins. diameter, stroke 55·1 ins.
Piston rod 8·1 ins. diameter.

$$\left.\begin{array}{l}\text{Gas pump diameter, 27·6 ins.}\\ \text{Air pump diameter, 31·4 ins.}\end{array}\right\}\text{Stroke 42·5 ins.; both pumps double acting.}$$

Fuel: producer gas from anthracite.
Date of test: September 1, 1904.
Duration of trials: Trial C = 1½ hours. Trial D = 1½ hours.

Results below are the mean of the two trials C and D.

Speed of engine	80 revs. per min.
Total IHP in power cylinder	856·8 metric HP
Brake HP	682·4 „ „
Indicated fluid resistance of pumps	89·2 „ „

$$\text{Mechanical efficiency } \frac{682·4}{856·8} = 0·79$$

$$\text{Proportion of pump resistance to total IHP } \frac{89·2}{856·8} = 0·104$$

Main piston speed	735 ft. per min.
Mean pressure on power piston	55·6 lbs. per sq. in.
ηp mean pressure equivalent of BHP	44·3 lbs. per sq. in.

A fuel consumption test was made with the same engine at its ordinary load which lasted from 6 P.M. October 31, to 5 P.M. November 1, 1904.

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[1] 'Gas Power,' by T. E. Junge, M.A., C.E., M.E. Hill Publishing Co., New York. 1908.

Assuming 78 per cent. mechanical efficiency, the mean brake HP developed during the run of 23 hours was 608, and the anthracite consumption in that time was 11,000 lbs.; the consumption per BHP hour was accordingly 0·787 lbs. per metric HP hour. As 1 metric HP = 0·986 English HP, the anthracite consumption was 0·8 lb. per English brake HP hour.

From the trials it appears that the maximum BHP of the engine was 682, and this power was developed with an ηp value of 44·3 lbs. per sq. in. and an actual mean pressure on the main piston of 55·61 lbs. per sq. in. The mechanical efficiency is somewhat low, practically 80 per cent. at full load; this is due to the somewhat high pump fluid resistance, which amounted to 10·4 per cent. of the total indicated HP. In an earlier test of the same engine the pump fluid resistance was 11·2 per cent. and the mechanical efficiency of the engine was 77 per cent. only.

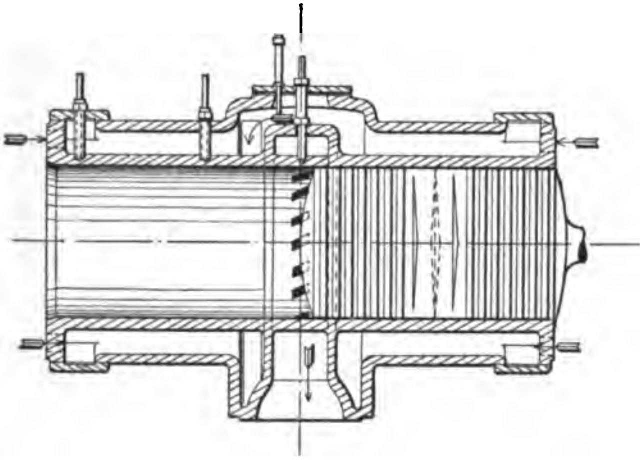

FIG. 151

At this stage of development the fluid resistances were high; later practice has greatly improved on this, as will be shown directly.

Mr. Junge calculates the mechanical efficiency of the engine in tests C and D as 89 per cent., and arrives at this result by deducting from the total IHP the pump resistances of 89·2, getting what he calls nett IHP in main cylinder as 767·6.

From this he calculates the M.E. as $\frac{682{\cdot}4}{767{\cdot}6} = 0{\cdot}89$.

This value, it is true, gives the efficiency of the mechanism apart from the fluid resistance, but it appears to the author to be misleading;

the English practice of calculating the M.E. to include fluid resistances is more accurate. By fluid resistance is meant charging and discharging resistances only; the compression resistance belongs to the thermodynamic cycle and is recovered in the main cylinder, so should be excluded.

The test is very creditable to the Koerting engine; 0·8 lb. per BHP is quite a favourable result with a producer of about 80 per cent. efficiency.

At the time of this test the Koerting cylinders were cast in one piece as to inner barrel and jacket casing, and the explosion and working stresses were divided between the two. Mr. Ernest Koerting made an interesting set of experiments on one of his cylinders to determine the temperatures attained in different parts of the metal.

Fig. 151 is a section of a 400 HP cylinder, showing the position of his thermometers. The metal of the cylinder was bored into at three positions, as shown, to within about 0·88 in. of the interior.

Tubes were screwed in and filled with mercury, into which the thermometer bulb was introduced, the bulb being well within the thickness of the cylinder walls.

Temperatures were read with the engine running under different loads.

Readings taken from 3.30 to 6.15 gave the following results:

TEMPERATURES OF THE INNER WALL OF A KOERTING ENGINE CYLINDER. (*Ernest Koerting*)

Time	3·30	4	4.15	4·30	4·45	5·0	6.15
Engine load . . .	¼	¼	½	⅝	¾	⅞	full
Temp. of cooling water discharge, degrees C. .	33°	35°	36°	37°	37°	37°	38°
Temp. of cylinder wall at:							
Outer point, degrees C..	75°	75°	83°	90°	88°	92°	94°
Second point, degrees C.	57°	57°	64°	66°	62°	63°	63°
Middle point, degrees C.	156°	157°	160°	165°	170°	170°	170°

Near the outer point the temperature rises to 94° C, at full load; at the second point the maximum is attained at ⅝ load, being 66° C., and in the unwatered metal carrying the exhaust ports it rises to 170° C. (middle point).

From his experiments Mr. Koerting considers that the mean temperature of the cylinder walls may be taken as 80° C., and that of the jacket casing as 29° C. And from this he infers that the expansion is insufficient to dangerously stress the metal even when the cylinder is cast in one piece. In the early engines, therefore, the working stresses were passed through jacket casing as well as working cylinder.

Further experience led to the abandonment of this practice, and in recent engines, while the explosion pressures are, of course, taken by the inner cylinder, it is divided circumferentially, so that the working stresses are taken by the jacket casing.

Recent Koerting engines by Messrs. Mather & Platt and other makers have been described by Mr. Alan E. L. Chorlton[1] in a paper entitled 'Large Gas Engines of the Two-cycle Type,' and this paper is important not only for its clear practical description, but from the fact that Mr. Chorlton is responsible for the later construction of Messrs. Mather & Platt's engines.

From that paper and information kindly given by the firm it appears that the main departures from early practice affect the arrangements of the main cylinder; the cylinder covers or heads; the valve gear; the air and gas pumps; the governor gear; and many constructive details. The cycle of operations is the same as already described.

First then as to the main cylinder. Fig. 152 is a longitudinal section of a recent 600 BHP Mather & Platt Koerting engine with cylinder heads and valves. Fig. 153 shows transverse and longitudinal sections of (*a*) a Nuremberg cylinder and (*b*) a Koerting cylinder.

Mr. Chorlton describes and discusses these as follows:

'*The Cylinder.*—Double-acting gas engine cylinders are broadly of two types:

Nuremburg .	. See (*a*) fig. 153
The Körting	. See (*b*) fig. 153

'In the first and well-known type the main feature is that all valve entrances are cast in one with the cylinder. The cylinder covers being of plain cylindrical jacketed form.

'In the second the cylinder is kept distinct in itself, and the covers carry the valve pockets.

'In the earlier days the greatest trouble was experienced with strains involving cracks round the valves; thus a crack in (*a*) type destroyed the cylinder, whereas in the second case a new cover only was required. Further, metal suitable for cylinder work is not always right for the high stress of combustion chamber duty; thus, for the first design the mixture is a compound one, but in the second case a suitable metal can be used for each duty. The author therefore prefers the second method, and as it was already adopted in the Körting engines, as first made by his firm, it was continued in use with modifications.

'In the older design in both cases no liner was used, the cylinder being cast in one piece, but at an early date the two-cycle manufacturers introduced liners for large sizes with successful results, with

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[1] 'Large Gas Engines of the Two-cycle Type.' Paper by Mr. A. E. L. Chorlton, M.I.Mech.E. The Manchester Association of Engineers, February 25, 1911.

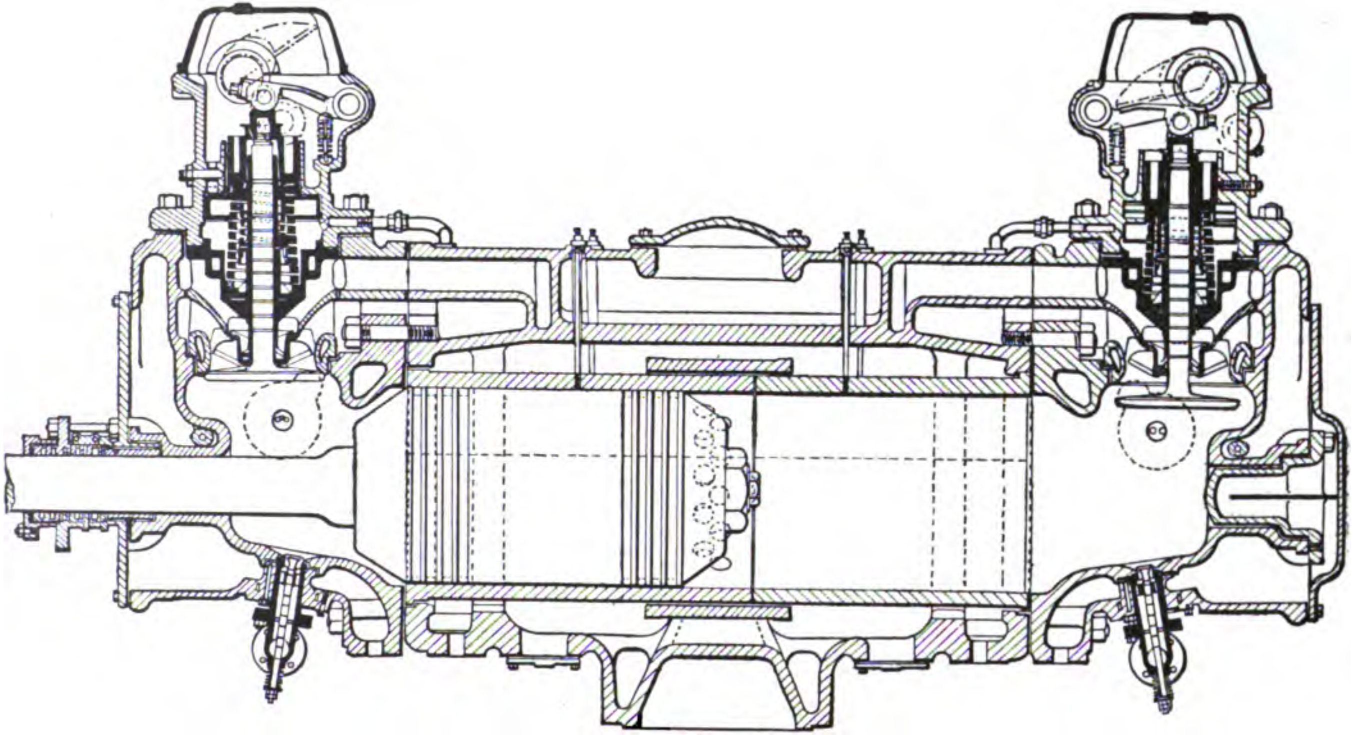

FIG. 152

the obvious advantage of easy replacement in case of wear. The four-cycle makers have now brought out a design on similar lines.

'In the design of cylinder shown in fig. 153 (*b*) it will be seen that a liner is fitted; it is in two parts, put in from both ends. In small cylinders the liner has not been found necessary, for apart from replacement from wear, considerations of strains set up in casting in the foundry are the main cause of its use in large engines of the two-cycle type. A gas engine cylinder when properly lubricated and using clean gas shows but little wear, a 400 BHP engine cylinder after one year's working showing an average wear of $\frac{1}{16}$ of an inch only; and this is by no means an exceptionally good result.

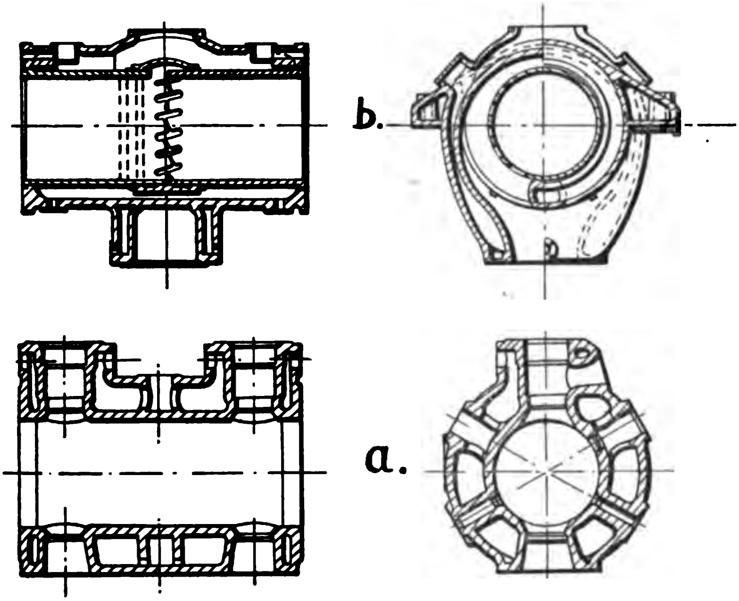

FIG. 153

'The cylinder has large hollow side brackets, arranged to give a pleasing external appearance; these carry within them the necessary gas and air passages conducting from the bed to the inlet valve at the top, and ends of the casting. These passages, starting at the faces of the brackets which bolt on to the main beams, curve and come out at either end of the cylinder to make joint with a similar facing on the cylinder cover when the latter is bolted on, as is shown in fig. 152.

'The metal of this cylinder is especially strong, as no consideration of internal rubbing surface has to be taken into account (the liner being of a different mixture). The exhaust outlet is carried through the swell beneath the cylinder with suitable cleaning doors on either side. The circulating water enters at the ends and leaves at the top of the cylinder, through the concealed return pipe, to the bed, so securing a clean and unencumbered outside appearance, with an absence of pipe work, so often made objectionable by a careless pipe fitter. The cylinder is secured to the beds by large bolts and the end thrust is taken up by large cross keys, the slots for which are machined in the beds by the same type of milling operation as the main bearings.

'It will be seen that the machining of the cylinder is a simple operation involving an ordinary boring machine with two double heads. The planing of the feet is also simple and is done on the same machine as the main beds. For such an operation it is easy to set dead parallel with the bore—a very important point. The cylinder liner used in large engines is a special hard metal divided circumferentially about the middle into two portions to allow for expansion. It is forced into the cylinder from both ends and is held in position by the covers, the exhaust ports being drilled in one half beforehand. The main studs for securing the cover are of special steel, fine threaded, and are put in by hand to ensure an even tightness. Machine studding whilst cheap is too risky, and at times tends to set up strains involving a possible crack, as owing to there being no accurate check on the drillers, the studs might sometimes be sent in too tight.

'The cylinder is lubricated from eight pipes by oil under pressure from an adjustable multiplunger oiler, a method that has worked well in practice, being much preferred to the one plunger with multiple feeds.

'*Cylinder Covers or Heads.*—The general shape of these is explained by fig. 152. In the larger sizes of engines they are now usually made of cast steel to obtain a higher factor of safety, and as additional precaution against irregular mixtures and strains in the iron foundry. The author believes that many of the troubles experienced in the past with cracked parts can be obviated by extra care being taken in the foundry both as to design and metal mixture used.

'From the design of the head it will be noted it is arranged to allow for expansion when cooling after casting, also when at work, thus avoiding internal strains. It is sometimes suggested that two-cycle engines, owing to the greater number of active strokes, involve greater temperature strains in the cylinder covers, but the author can hardly agree to such a view. The number of stresses also depends upon the speed, and the two-cycle engines are usually run slowly; further, the temperature variations are distributed in a better manner than those of the four-cycle type, where the inlet and outlet valves are at the same end of the cylinder. The "Gleichstrom" steam engine now being largely introduced, which works on a similar principle to the two-stroke cycle gas engine, has a similar temperature gradient.

'Ignition plugs are fitted in the side of the cylinder covers in duplicate placed directly opposite to each other, thus incidentally allowing of testing in position. This particular design was adopted some four years ago and has proved very successful in practice.

'From the constructional point of view it is advantageous to make the cylinder cover fit either front or back end, and be in fact good for either engine. This is secured in this design with the opposite situated plugs, the facing carrying the ignition bracket being used on the other side for the cylinder oiler; the air starting non-return valve is situated on the centre line below. The cylinder cover is carried up at the top as shown, in order to form a chamber for the inlet valve and its removable seat, and this portion is shaped at the end, and linable with the main cover joint. Thus at one operation the joint is made both for main cylinder and gas and air passages.'

From fig. 152 it will also be observed that the exhaust gases do not discharge all round the cylinder as formerly (see fig. 147), but pass out at opposing sides into chambers which unite below into one chamber at the bottom. By this arrangement there are no exhaust apertures in the liner either top or bottom, so that oil discharge at the bottom ports is avoided and cylinder oil is economised; further, water surrounds the liners and gets free access at every part except that necessary for the side exhaust chambers. This is an excellent arrangement. The exhaust, too, passes through bored holes instead of by way of slots, and all these holes are bored in one end of the liner. The separated liner is efficiently held at the outer ends, and it is pressed into the central bored ring sufficiently tightly to prevent leakage of water into the exhaust pipe, but still free enough to allow a slight longitudinal movèment for expansion. Here it will be observed the working stresses are carried through the water casing and the large saddle piece. Below, at the ends, are seen the compressed air valves for starting, and under the inlet valves are placed low-tension make and break igniting plugs, indicated diagrammatically.

With regard to valves and valve gearing Mr. Chorlton says:

'In all double-acting gas engines, whether four- or two-cycle, a side or lay shaft, driven from the main crankshaft by screw or bevel gearing, is the usual method of operating the valves. Such a general arrangement for double-acting engines was formed from that of the horizontal drop valve type of steam engine as made by Messrs. Sulzer & Co., and was very common on the Continent.

'The early Körting engines built in England were on this model,

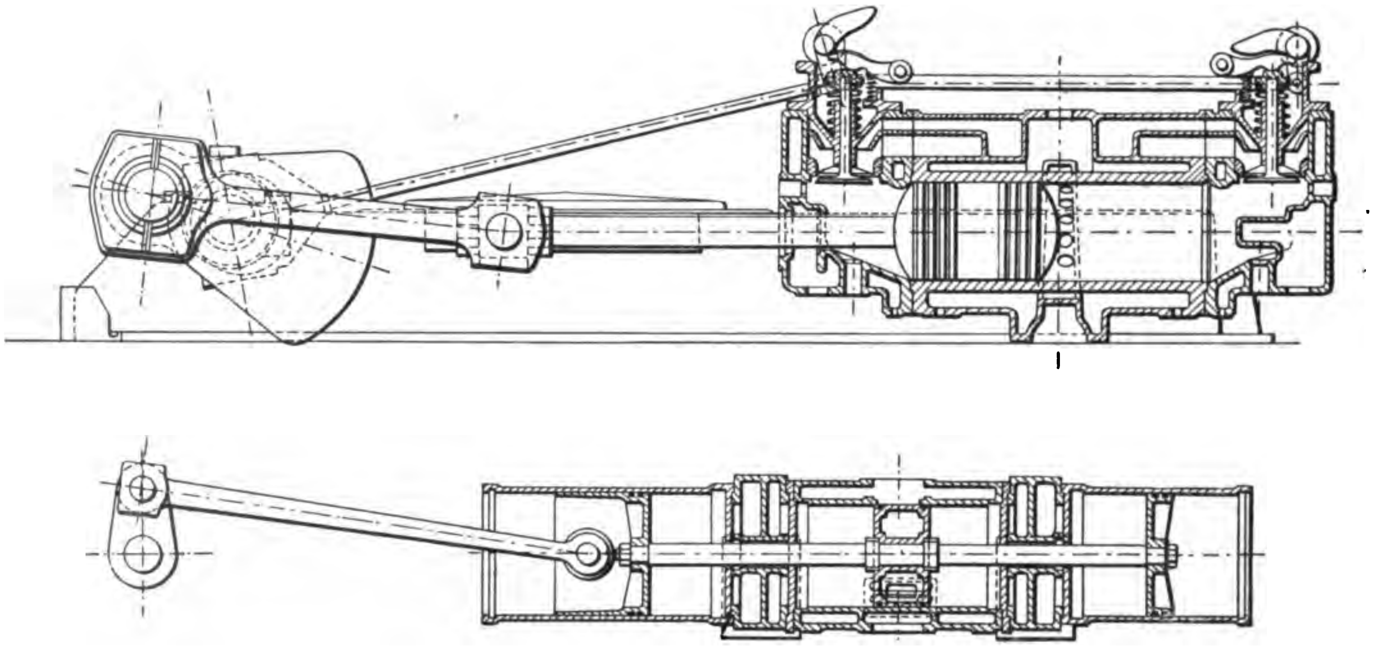

FIG. 154

and it was from their running that certain conclusions were drawn, which, though made from a two-cycle type, are yet common to all types of gas engines. The objections to the cam shaft form of valve gear were :

'1. Noise of the driving gearing.

'2. Unequal wear due to sudden action of cams.

'3. Bad effect on governing, if the governor is driven from it, for same reason.

'4. Torsional spring of side shaft.

'5. Cost, upkeep, and renewals.

'Dealing first with the governing, it seemed that to get the very best effect for running alternators in parallel, and to produce, for mill driving, the Moscrop line, so dear to the Lancashire manufacturer, that the governor must be driven direct from some steady rotational part such as the crank shaft, and that driving from the side shaft and its irregular angular movement must be avoided.

'As the engine is of the two-cycle type no awkward inaccessible exhaust valves have to be operated, and one valve stroke only for each power stroke is required, and that only to operate the admission valve. The possibility of greatly simplifying the whole valve drive was apparent, and resulted in the evolution of the modified gear shown in fig. 154. Such gear, with practically no alterations, has been used ever since it was first put to work five years ago. The whole arrangement, incidentally, is of course cheaper than the old side shaft type, to which it is much preferable as having fewer parts, and generally being quieter and better in action.

'It be will seen that it consists of a simple eccentric on the crankshaft driving direct through a rod on to a pair of rolling levers on the front inlet valve, the back inlet valves being driven from these by a connecting-rod, as shown in the figure. These levers are made in one piece of cast steel, for owing to the nature of their duty it was feared that the built-up type with keys would ultimately work loose. They have run with great success and show but little wear since the first.

'The inlet valves themselves are of the plain mushroom type, operated direct through the "crocodile" levers as indicated, and have a suitable air dashpot to ease their return on to the seat. The whole arrangement of valve cage, seating, dashpot, &c., is indicated in fig. 152. It is a simple gear and is generally as employed in ordinary gas-engine practice. It does not occasion any special work in tooling, being easily dealt with on a combination turret lathe with accuracy and at economical rates. No special arrangements for grinding in are necessary, as it is a common fact that this valve rarely indeed requires re-grinding, running quite satisfactorily for years. The inlet valve is common to both gas and air, a very peculiar charging cycle, controlled by the gas and air pumps, making this perfectly satisfactory for all ordinary duties and retaining at the same time the utmost simplicity. For possible cases where higher efficiencies with rich gas may be required, and to meet occasional demands, an additional gas valve is sometimes fitted, but this is really hardly worth while, for it reduces the simplicity, and if economies are to be made experience has shown that such should not be attained by reducing the simplicity in the engine, a fact which has been kept in mind throughout the whole of this design.'

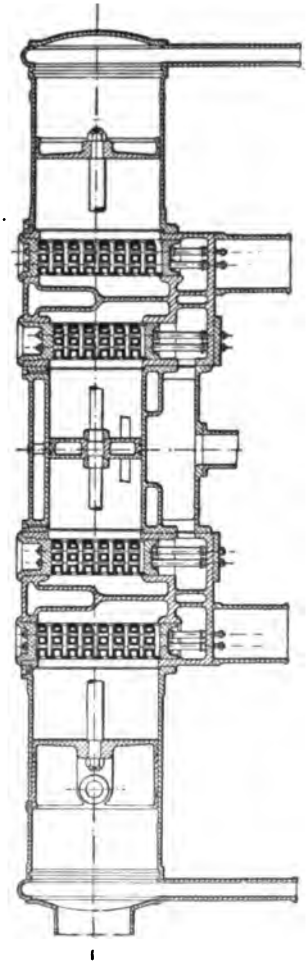

FIG. 155

A most important modification is found in the arrangement of the charging pumps. It will be observed at fig. 150 that owing to the arrangement of air pump at one end and gas pump at the other the air passage leading to the main inlet valve nearest to it is shorter than that leading to the far end of the main cylinder. This is also true of the gas passages. The clearance between the inlet valve at one end of the cylinder and the air or gas pump is thus less than at the other. This is a disadvantage and leads to variation in charging one end of the main cylinder as compared with the other. The piston valves of both pumps also cause increased expense of construction. These difficulties of lack of symmetry in air and gas passages are obviated very cleverly by Mr. Chorlton's design of air and gas pump shown in section at fig. 155, in which a double-acting gas pump is placed centrally between two single-acting air pumps. All three pistons are driven by the same piston rod, and the air piston at the left-hand side is of the open trunk type directly operated by a connecting-rod. This cylinder serves instead of the older crosshead slide. The pumps are supplied and discharged through numerous small automatic lift valves placed in cylindrical cages which can be removed vertically for cleaning. The gas piston moves for more than the first half of each stroke open to a longitudinal slot in the cylinder side, which allows the gas to be returned to the inlet passage or chamber.

In Mr. Chorlton's paper the pump arrangements are thus described :

'Alongside the power cylinder are placed the charging pumps for air and gas ; the action of these is something similar to that of the original "Körting" type, though the arrangement is different, and the whole combination is simplified. The double-acting pump for the gas is placed between the two single-acting pumps for the air. The three pumps are thus in line and secured to one another as well as to the engine frame. The pump cylinders are separated from each other by intermediate valve boxes, each of which is divided by a diaphragm into passages for the gas and air respectively, and is also provided with automatic valves, now made easily removable.

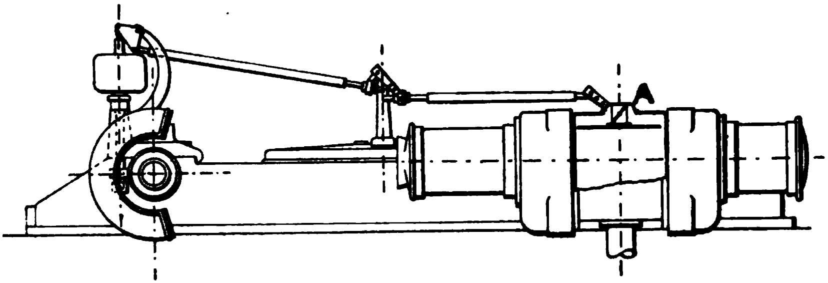

FIG. 156

'Fig. 155 shows a section of these pumps and valves.

'With the object of delaying the charging of the gas until after the scavenging by air has been effected, ports are provided about the middle of the gas pump, through which the gas passes back to the suction side during the first part of the stroke, and thus delivery is delayed until the scavenging has been performed by the air pump.

'When the cylinder pressure falls to atmosphere, the admission valve is opened, and a scavenging charge of pure air is sent into the cylinder by the air pump, effectually clearing out the products of combustion. The gas pump (the discharge of which, as mentioned, takes place later than that of the air pump) now sends a supply of gas into the cylinder, which discharge, effectively mixing above the inlet valve with that of the air pump, forms the combustible mixture necessary for the power stroke. It must be realised that the charge is of a known quality, and is put or measured in by pump piston displacement, so as to effectually prevent any loss through the exhaust ports. The exhaust ports are closed by the return stroke of the power piston during the charging, and the mixture is then compressed, ignited electrically at two distant points, and the impulse transmitted to the piston in the usual way. The same cycle is repeated at each end of the piston alternately, the air pump sweeping all burnt products through the exhaust ports, and thus preventing the pre-ignition of the incoming charge by contact with hot gases which would otherwise remain in the cylinders. This thorough sweeping out of the burnt products by the scavenging charge is an undoubted advantage and is conducive to good running and the prevention of wear. On an examination of cylinders, after months of steady running, it has been proved beyond doubt the ports are kept particularly clean and no undue wear takes place on the metal.'

The governing is also simplified as shown at fig. 156. It is thus discussed in the paper:

'The standard design is as shown in the sketch, fig. 156. Its essence is simplicity, and the author is quite willing to admit that as regards pump efficiency a more theoretical arrangement of the trip, or variable cut-off type (such as is sometimes fitted), is quite possible, but such involves considerable complications and an increase in number of parts liable to wear or "tar up," and cannot in his opinion secure any materially better governing than the really simple arrangement used. In principle this is a by-pass governor; the movable wing valve A controlled from the governor (direct driven from the crankshaft) allowing more or less of the charge in the gas pump to be by-passed to the other end, thus regulates the amount going forward to the main cylinder, according to load.

'This governing is extremely effective, for acting as it does on the charge stroke of the pump, it is but one stroke off the actual ignition in the main cylinder, whereas all four-cycle engines are two strokes off owing to the previous suction of the charge being the governed stroke.

'In the particular engine under description similar wing valves on the air pumps control the air for the mixture (the scavenge is a constant), and they can be varied with respect to each other at will. Usually these valves are not necessary and are not fitted in the smaller engines. In large engines they ensure a certain firing at both ends of the cylinder at dead-light load, and so facilitate running and synchronising for alternator work.'

The ignition is of the low-tension internal make and break type, having a trip magneto operated from a reciprocating plate carrying trip pieces which catch and move the armature, which is released by a wedge piece.

The cranks are separate from the crankshaft, the crank webs being

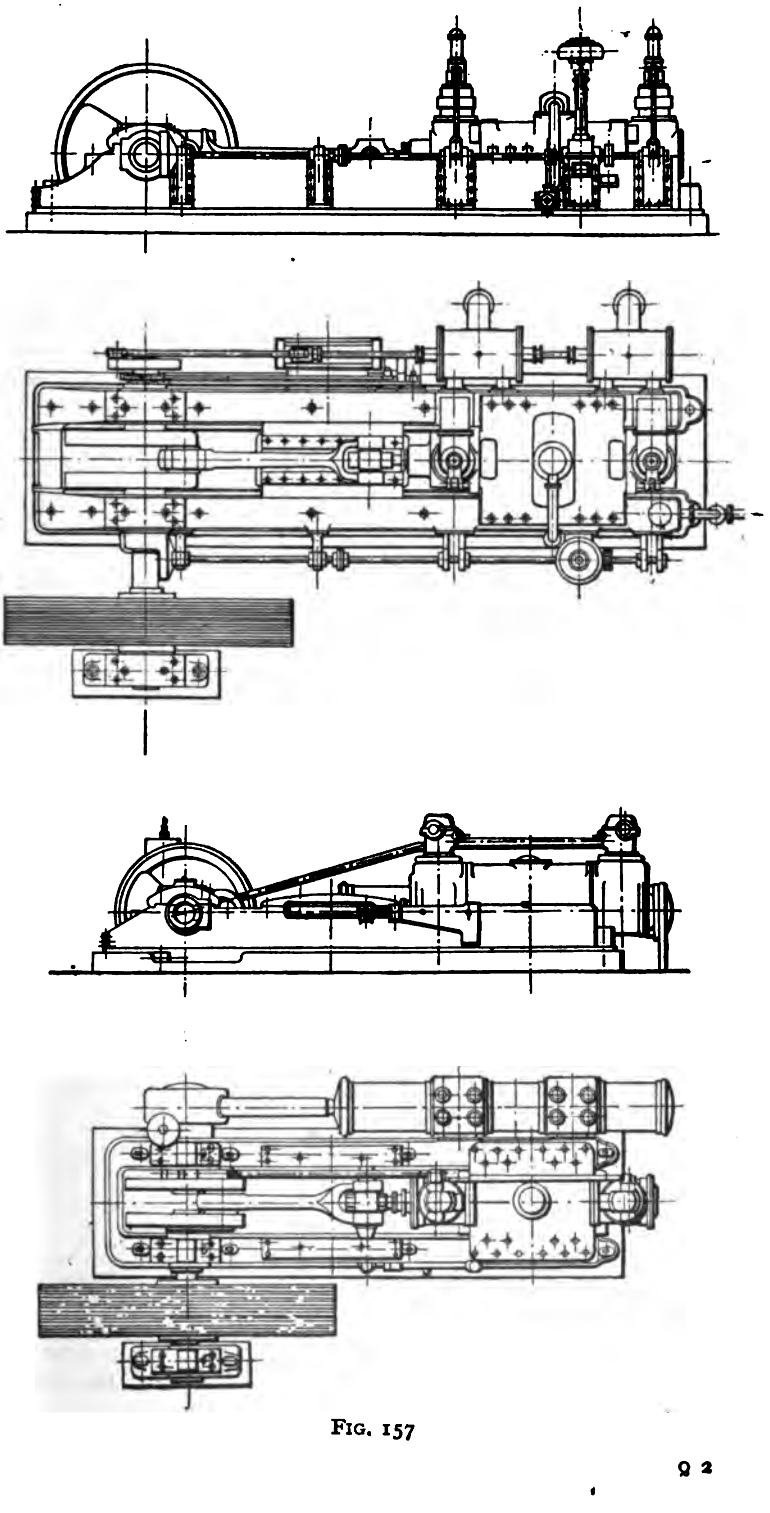

FIG. 157

each in one piece with its opposing balance weight. The cranks are shrunk on the shaft and on to the crank pin.

The whole effect of the changes can be seen at a glance by means of fig. 157, which shows an elevation and plan of the German design above and the English below.

Figs. 158 and 159 are from photographs of the early German and the recent English designs.

The great simplicity of the English design is apparent. The absence of the side shaft with its driving gear and levers, the simplified governing, and the symmetrical arrangement of central gas pump between single-acting air pumps combine to make an engine of more sightly appearance as well as of more economical construction.

Mr. Chorlton gives examples of indicator diagrams which show that with a mean pressure of 62 lbs. per sq. in. in the main cylinder the pump fluid resistance amounted to 5·24 per cent., or only 3·2 lbs. per sq. in. referred to the main piston area and stroke, and this includes the suction resistance of a gas producer. This engine ran at 112 revolutions per minute driving a mill.

In the case of another engine indicating 1080 HP at 76 revolutions per minute the pump resistance was 5·97 per cent. of the IHP, and the mechanical efficiency in the English sense was 82·5 per cent.

These results are very satisfactory and show a great improvement on the 10 and 11 per cent. fluid resistances of the early German engine.

A recent test of an engine of 600 BHP shows the consumption of anthracite to be under 0·8 lb. per BHP hour at full load.

Messrs. Mather & Platt give the following dimensions and particulars of their 1902 engines:

SOME DIMENSIONS AND PARTICULARS OF KOERTING ENGINES. (*Messrs. Mather & Platt.* 1902)

Brake horse-power	Cylinder diameter	Stroke	Revs. per min.	Piston speed, feet per min.	Approximate weight *
400	22½ ins.	39½ ins.	110	725	50 tons
500	25 ins.	43¼ ins.	100	721	75 tons
600	27½ ins.	48 ins.	90	720	100 tons
700	29½ ins.	51¼ ins.	86	735	127 tons
1000	37⅜ ins.	63 ins.	70	735	190 tons

* Weights include flywheel and outer bearing.

For this rated power of 400, 500, 600, and 700 horse the value of ηp may be taken as nearly 47 lbs. per sq. in., and if the mechanical efficiency be taken as 80 per cent., then the actual mean pressure on the piston would be nearly 60 lbs. per sq. in. For the 1000 HP

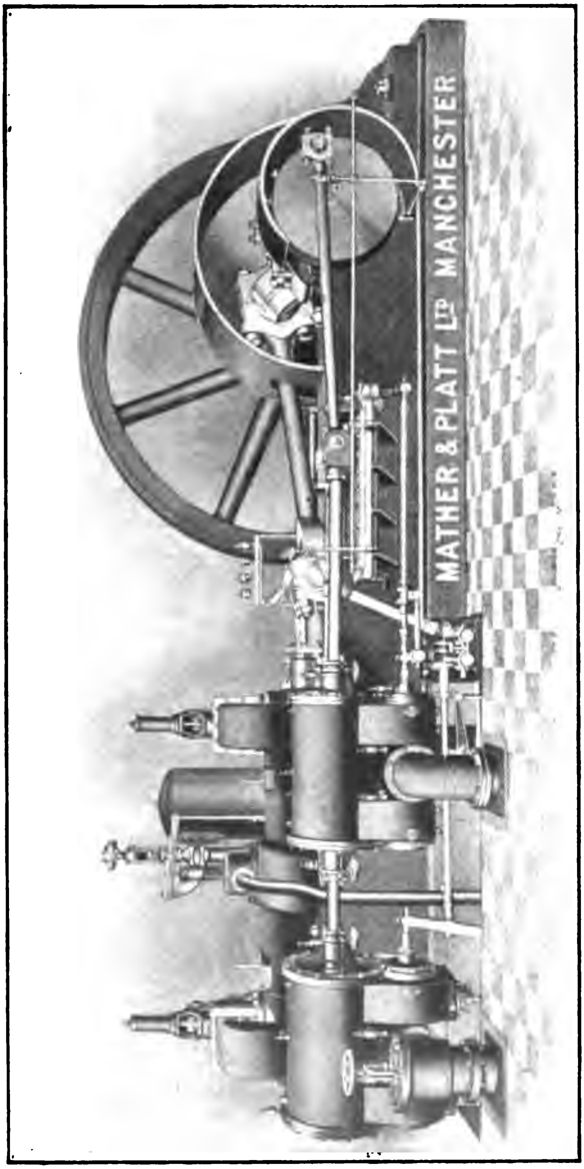

Fig. 158

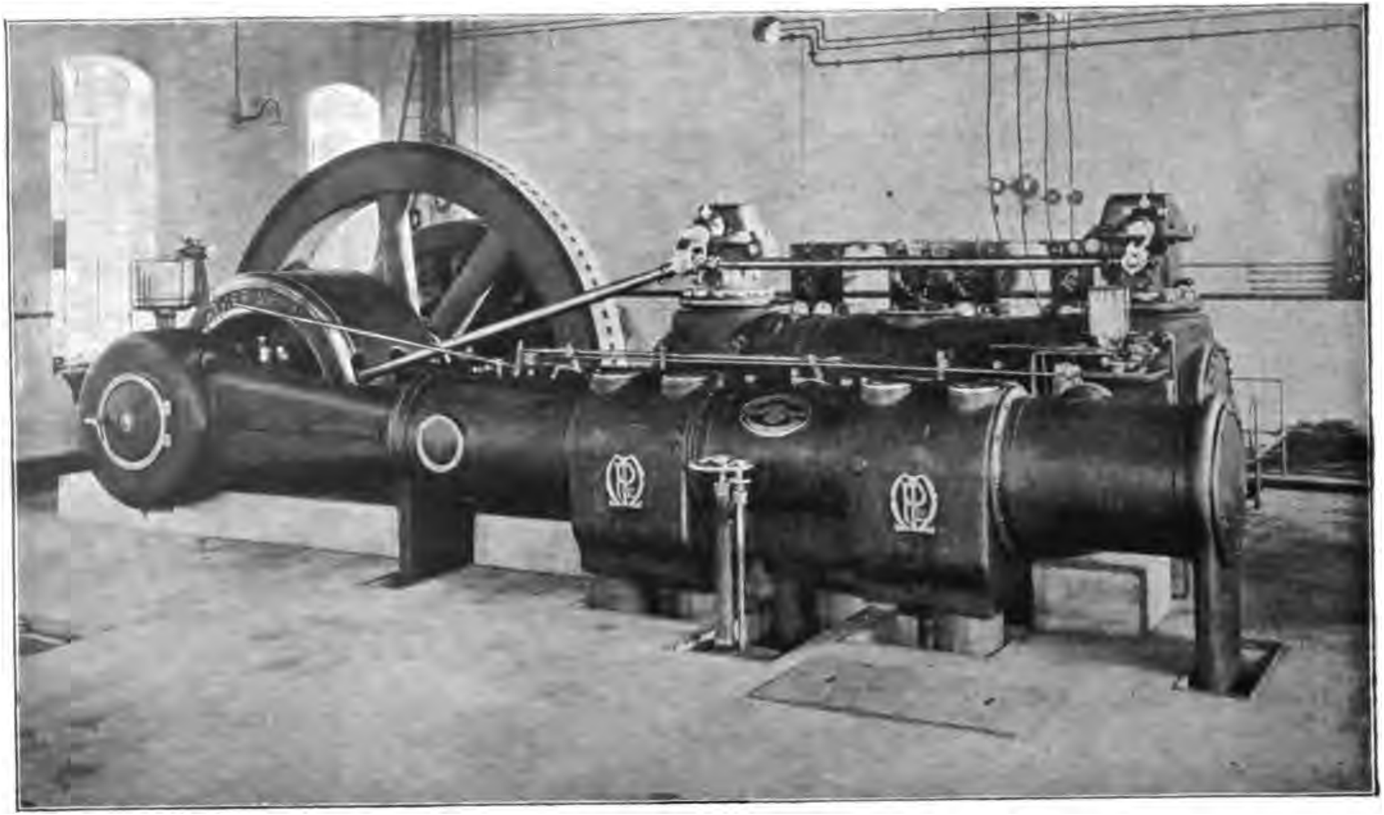

FIG. 159

engine the valve of ηp is 41 lbs. per sq. in., and actual mean pressure on the piston on the same assumption is 51·3 lbs. per sq. inch.

Messrs. Mather & Platt give the following particulars of their more recent engines, which show very material reduction of weight per power unit :

POWER AND WEIGHT OF RECENT MATHER & PLATT KOERTING ENGINES.
(*Messrs. Mather & Platt.* 1908)

Brake horse-power	Revs. per min.	Approximate net * weight engine	Weight of flywheel
400	100	45 tons.	10 tons.
500	110	50 tons.	13 tons.
600	100	75 tons.	16 tons.
700	95	85 tons.	20 tons.
1000	80	135 tons.	32 tons.

* Weights include flywheel and outer bearing.

The recent engines in some cases thus show as much as 40 per cent. reduction of weight compared with the 1902 type.

Messrs. Mather & Platt have applied these engines successfully to the generation of electrical power, blast furnace blowers, direct driven pumps, and mill driving; in all they have installed about 35,000 HP.

Among the builders of the Koerting type engines on the Continent may be mentioned the Siegener Maschinenbau Siegen ; the Ascherslebener Machinenbau A. G. ; Gutehoffnungshütte, Oberhausen ; and Gebr. Klein Dahlbruch ; and in America the De La Vergne Machine Co., New York. The Siegener Co. have built engines giving 2000 HP in a single cylinder. Mr. Chorlton gives the following as the principal dimensions :

Stroke	1400 mm. (about 4 ft. 7 ins.)
Power cylinder diameter	1100 mm. (about 3 ft. 7 ins.)
Blowing " "	2550 mm. (8 ft. 4 ins.)
Speed	30–90 revs. per minute
Normal load	1000 cub. m. (35,316 cub. ft. at pressure of 0·8 atmos. (11·76 lbs. per sq. in.)

The inlet valves are very large : 19·68 ins. diameter with a lift of 3·14 ins. ; they run quietly at 90 strokes per minute. They are driven from cranks on a side shaft, through rolling levers, and they are closed by vacuum piston as well as by spring.

Early in 1909 Gebruder Klein installed four Koerting two-cycle engines of 1050 BHP per cylinder at the works of the Frodingham Steel & Iron Co., Ltd.

The power cylinder is 35½ ins. diameter with a stroke of 55⅛ ins., and the speed is 70 revolutions per minute. Blowing cylinders are attached to the main piston rods and the engines are actuated by gas from three blast furnaces.

In America the largest installation of Koerting type engines by the De La Vergne Machine Co. of New York is found at the Lackawanna Steel Works, Buffalo, amounting in all to 40,000 BHP.

Messrs. Koerting and their licensees are to be congratulated on their success in competing with the four-cycle engines in the field of the large gas engine. In the author's view the two-cycle engines can still be greatly improved, and in the near future they will secure a much larger representation among large gas engines for all purposes.

THE OECHELHAUSER TWO-CYCLE ENGINE

One other two-cycle engine of importance has also been produced in Germany by Dr. Oechelhauser; an early example was inspected by the author at Hoerde in 1902. Several of these engines were at work at the Hoerde Iron Works in that year. Two of the engines were twin cylinder motors of 600 HP, and one was a single cylinder motor of 500 HP. The earliest of these motors was of the twin cylinder type of 600 HP and it was set to work in 1899, and had been at work three years when inspected by the author.

Fig. 160 is a diagrammatic plan of this engine.

In this two-cycle motor there are two working pistons, which move oppositely in one working cylinder which is open at both ends. The pistons are single acting. These pistons directly control both the discharge exhaust gases and the admission of the air and gas to form the explosive mixture to be compressed.

When the pistons have travelled full in, or nearly full in, and the mixture of gas and air is compressed between them, it is ignited by an electric spark and the working stroke takes place. The pistons are driven apart by the expanding gases on this stroke.

The main shaft has three cranks. The centre crank operates the front piston by means of a connecting-rod; the two side cranks operate crosshead slides which are placed at the sides of the main cylinder, while rods from these slides operate a large crosshead behind the engine, and from this crosshead is operated the back piston and also the blowing piston.

In some cases the air and gas pump is placed beneath the engine in a pit and is operated by means of a rocking lever connected to one end of the back crosshead; in other cases the pump is actuated directly from the back crosshead by a piston rod as shown on fig. 160.

The action of the engine is as follows, referring to the modification shown at fig. 160. When the back piston moves it operates the double-acting pump piston, which on one side serves as air pump and on the other as gas pump; gas and air are separately pumped into separate reservoirs at a constant pressure of about 0·3 to 0·4 atmosphere.

The motor pistons separate, and towards the ends of their strokes one piston overruns the exhaust ports and discharges the contents of the cylinder down to atmospheric pressure. When the pressure has reached atmosphere, the other piston overruns first the air ports, when air is at once delivered into the cylinder from the air reservoir. This displaces some of the exhaust gases. The further movement of the same piston uncovers the gas ports, and gas is then delivered from the gas reservoir into the cylinder with the continuing air flow. It thus mixes with the ingoing air and forms an inflammable mixture within the cylinder. By this arrangement no gas and air mixture is formed in chambers or passages — explosive mixture is formed only within the cylinder. As the engine rotates, the pistons cover the respective ports and the charge is compressed. In order to prevent the escape of unconsumed mixture of gas and air through the exhaust ports the dimensions of the working cylinder are so

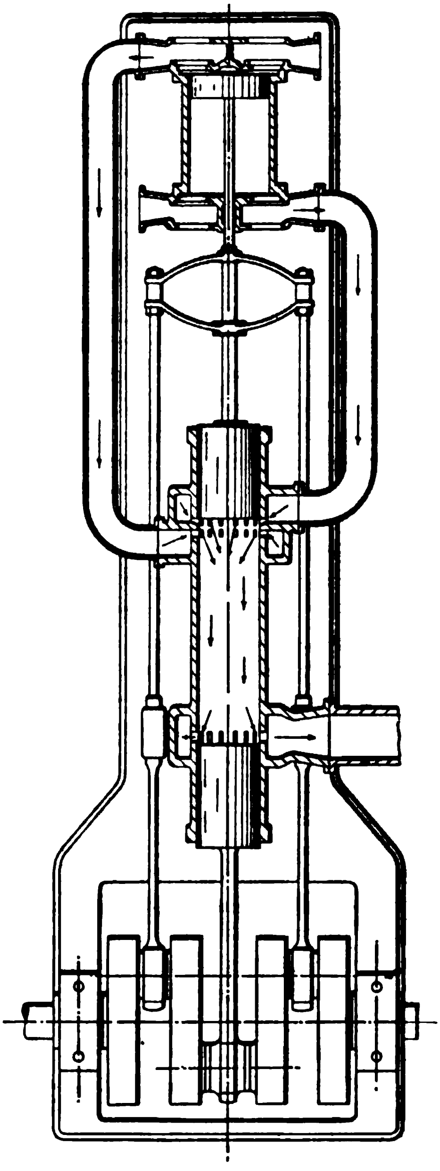
FIG. 160

proportioned that the greatest quantity of gas and air discharged into the cylinder amounts to only about 70 per cent. by volume of the cylinder capacity. In this way a minimum of charge loss is incurred at the exhaust ports.

Electric ignition is used ; in the early engines two magneto igniters were operated simultaneously and actuated sparking points placed at opposite sides of the cylinder, and so ignition was rendered certain and sufficiently rapid.

Governing is effected, as in the Koerting engine, by reducing the gas charge.

So far as the cylinders are concerned, the valve operations are of the simplest possible kind. One of the greatest difficulties which gas-engine builders have to face in dealing with blast furnace gases is that of getting gas free from dust. The Oechelhauser engine has been specially designed to meet this difficulty, and no doubt the smooth plain cylinder—smooth from end to end without ports or irregularities of any kind—aids materially in preventing the accumulation of dust which may find its way into the engine cylinder. It is a great advantage also that all sensitive valve organs are removed from the cylinder.

The modern plant for cleaning blast furnace gas, however, is so perfect that this difficulty is felt but little to-day, and it is now quite permissible to utilise lift valves, as is shown by the success of four-cycle engines for large powers.

The system has other advantages ; although three cranks are employed, yet the mode of operation allows the two pistons to be largely balanced one against the other, and as the thrust of the explosion and expansion is taken between the pistons and transmitted to the opposing crank pins, but little of the driving stresses reach the engine bed. The engine bed becomes thus relatively light.

Dr. Oechelhauser claims that the first large unit engine 600 HP in twin cylinders put to work by his company at Hoerde Iron Works in 1899 preceded the large power engines of other makers. This engine had two cylinders each of 18⅞ ins. diameter and equal strokes of 31½ ins. for all four pistons, running at 135 revolutions per minute. This engine was started in 1898 and taken over by the purchasers in January 1899.

The Deutsche Kraft-gas Gessellschaft was formed in Berlin to exploit this type of engine, and the principal continental licensees were Messrs. Borsig and Messrs. Aschersleben M. A. G. The total horse-power supplied by these firms is over 40,000.

Messrs. Wm. Beardmore & Co., Ltd., of Glasgow became licensees in 1904, and have built, from that date to the beginning of 1910, 28 engines of 26,000 BHP total.

Fig. 161 shows an elevation, a plan, and an end elevation of a double cylinder 2000 HP set of Oechelhauser engines as built by Messrs. Beardmore in 1906. It will be seen that two separate 1000 HP engines are arranged side by side with the dynamo between them. The air and gas pumps are placed in a pit below the engine and driven from the back crosshead slide by a connecting link operating a lever which actuates a cross shaft. This cross shaft carries levers at its ends, one of which operates the air pump and the other the gas pump. The air and gas pumps are easily distinguished in the drawings by their difference in diameter. The gas pump is much the smaller of the two.

The cylinder is of the built-up type and is free to expand at its ends through packing glands in the surrounding water jacket casing.

The governing is effected by by-passing the gas; it is thus described by Messrs. Beardmore:

'A rod is actuated by a cam on the lay shaft, and raises the by-pass valve on the gas delivery pipe. The length of time this by-pass valve is held open, and consequently the amount of gas left to be compressed into the reservoir, is controlled by the governor through an eccentric pin. A similar by-pass valve is fitted on the air system.'

Ignition was accomplished on the Lodge system, described later in this work (see Chap. III.).

In an interesting paper [1] read by Messrs. Stokes and Cunningham on November 11, 1909, the authors—who are engaged at Messrs. Beardmore's in the design of these engines—describe the later and improved form to which these engines have developed in Scotland.

Messrs. Stokes and Cunningham thus describe the differences:

'Fig. 162 shows a comparison of a former design with two of the latest arrangements, lettered respectively A and B. The difference between A and B consists solely in the position of the charging pump. In A the pump is driven off the back crosshead in a similar manner to that adopted in the first Oechelhauser engines, while in B it is driven by a crank disc on the end of the crankshaft. Each arrangement has special advantages of its own, and the one to be adopted can only be decided upon after due consideration of the conditions obtaining in any particular case.

'It will be seen from a survey of the new design that all the advantages have been retained and the objections overcome as follows:

'(1) Reduction in length. For a given power of engine the overall length has been considerably reduced, i.e. by 25 per cent. and 35

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[1] 'The Oechelhauser Gas Engine in Great Britain,' by Jas. W. B. Stokes and James Cunningham, Wh.Sc. Glasgow University Engineering Society, Nov. 11, 1909.

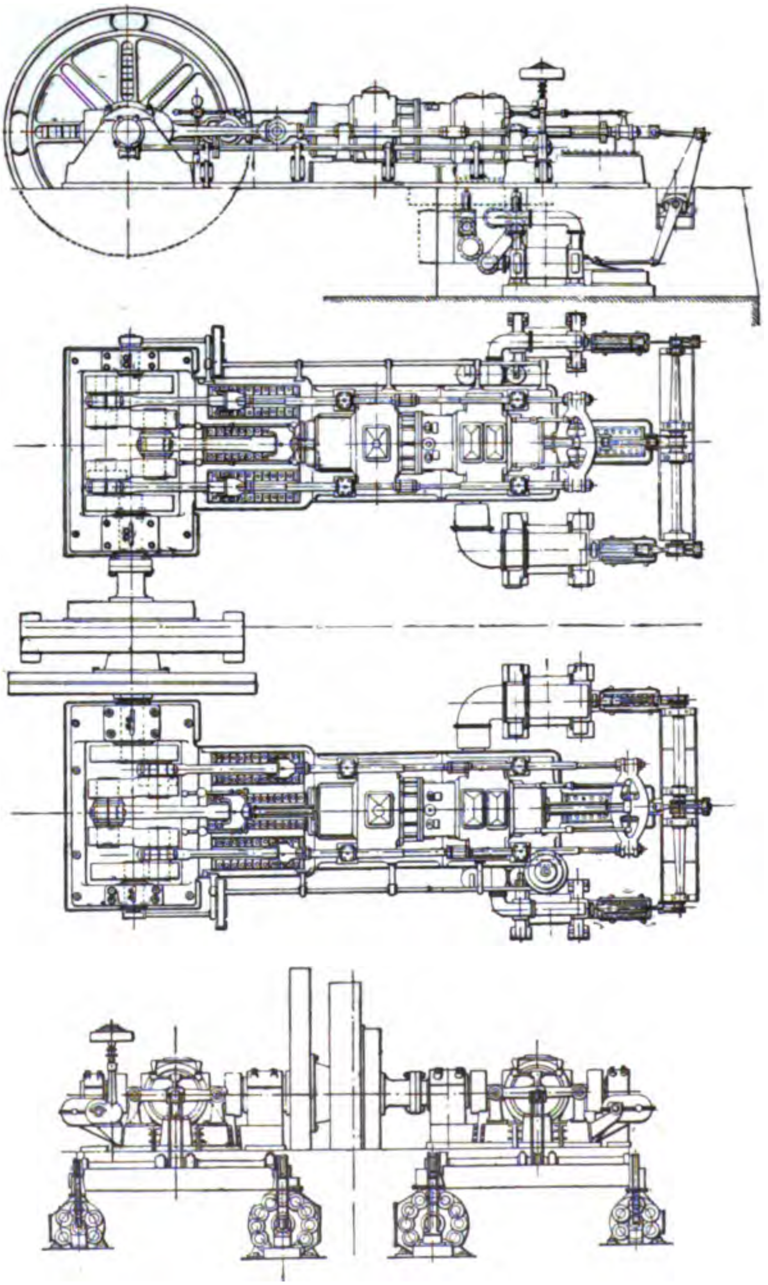


FIG. 161

per cent. respectively in the two arrangements A and B of Fig. 162, as compared with the engine of equal power of the older design shown at the top of the same figure. This has been accomplished mainly by a noticeable reduction in the length of the working cylinder, which has now piston rings fixed at each end so that they ride on the pistons and prevent escape of gases from the receivers. Thus the pistons are enabled to overrun the cylinder a considerable portion of their stroke after the manner of plunger pumps as commonly seen. Any fear as to a possible whipping action of the pistons protruding from the cylinder is removed by the knowledge that they are firmly secured to the crossheads, which are guided in position both vertically and horizontally. By this design the production of cylinder liners and combustion chamber in one continuous piece of metal is rendered possible, avoiding the joints otherwise necessary, while the liner may also be withdrawn from the casings apart from any further dismantling of the engine.

'(2) Economy in construction has been effected by this design. A considerable reduction has been made in the weight of the whole engine, as would be expected from fig. 162. The dimensions of the receiving chambers for the air and gas charges and for the exhaust have been increased, so that these now support the working cylinder from the foundation without intervening sole-plates. This arrangement of the receiving chambers thus stores the charges in the closest proximity to the inlet ports, so that during the admission period they have the shortest distance to travel, and enter free from pulsations. This leads to a reduction in the necessary port area, and consequently an increase in the effective stroke. The temporary stratification aimed at with the scavenging and mixture charges is more uniformly attained, hence a larger quantity of combustible may be retained in the cylinder. The design thus tends to give a higher mean effective pressure and to develop a higher power for a given size of cylinder. This leads to further considerable economies.

'(3) The design of three-throw built-up crankshaft has been subjected to a thorough calculation of strength by Professor Eugen Meyer of Charlottenburg. The result of these calculations was to show that the stresses in the shaft, even with premature ignitions, are well within those permissible with ordinary Siemens-Martin steel, of which these shafts were built, whilst the stiffness left nothing to be desired as compared with the crankshafts of other engines.

'These shafts have now been in operation for years without showing any signs of weakness in the parts calculated. As a matter of fact, on the first engine built in this country, which was employed in a rolling mill, trouble was experienced with the shaft. This was due to the method of fixing the crank pins, which in this case were simply

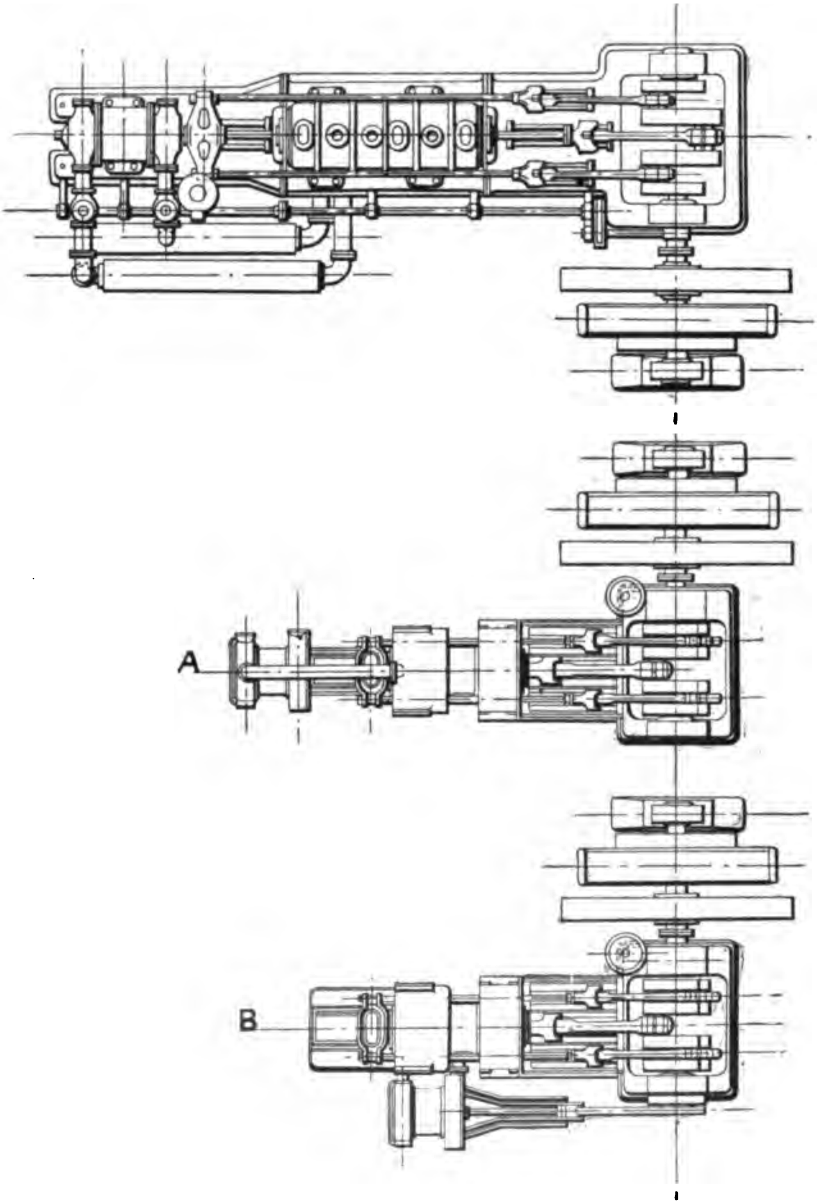


FIG. 162

pressed into the webs with the usual dowel pins. This method of construction proved unsatisfactory, and all other shafts were made by shrinking the webs on the pins.

'Recently we have taken advantage of the special properties of the fluid compressed steel manufactured by Messrs. Beardmore to design the shafts for our engines as solid forgings, and at the same time to reduce the crank pin dimensions, a reduction found possible with the very efficient lubrication now provided. By these means the weights and manufacturing costs of the shafts have been considerably reduced without in any way decreasing their suitability.

'(4) Dealing with the disadvantage of limited speed, in the new design the length and consequently the weight of the side rods and also of the back crosshead have been reduced, with a resultant decrease in the inertia forces due to these reciprocating masses. The speed of rotation of the engine may thus be safely increased without stress reversal in the side rods. This principle of the reduction of the reciprocating masses attached to the side rods is carried further in the second arrangement (B) than in the first (A), by the removal of the pump from the back to the side of the engine, permitting a still higher speed to be attained.

'(5) Concerning the pulsating effect of the open-ended cylinder on the engine room atmosphere, we frankly admit the new design shows nothing to mitigate this. In this connection we may point out that with a two-line or coupled engine, such as would undoubtedly be employed for alternator work and large powers, the pulsations are minimised by the compensating action of the pistons, as is also the case with an engine of tandem design.

'(6) In connection with the objection as to the low mechanical efficiency of the Oechelhauser engine, we are aware that in our earlier engines the defect has been largely due to the high percentage of the indicated horse-power absorbed at the pumps, amounting to about 25 per cent. of the total developed at approximately full load. This bad result was, however, largely caused by defects in the mechanical arrangements, as some slight alterations on the pumps of these particular engines reduced the ratio to 10 per cent. at similar loads.

'This latter figure has since been improved upon during an official test of a 1000 BHP engine developing 1200 IHP when the pump work was ascertained not to exceed 7 per cent.

'We mentioned when dealing with economy in construction that the gas and air charges are stored in close proximity to the inlet ports in symmetrical receivers surrounding the cylinders. By this means an important reduction in the pump work is attained, and we may, therefore, conclude that the figure of 7 per cent. obtained with the old economical design will be still further improved upon, while the loss in friction will decrease with the reduction in the weight of the moving parts.'

Fig. 163 shows in plan (part section) and end elevation the later form of Beardmore Oechelhauser engine, from which it will be seen that the cylinder is cast in one with the water-jacket casing; that it is greatly shortened so that the length between the outer ports and the open end is only half the length of the stroke. In the old arrangement, fig. 160, it had to be more than the length of the stroke at both ends. By the use of fixed external rings and plunger pistons, coming out of the cylinder immediately the working strokes begin, a great saving of length is effected which permits reduction of length of the side rods, although the lengths of the pistons themselves are slightly increased.

This design is without doubt a considerable improvement on the earlier forms.

Diagrams and Gas Consumption.—After discussing the questions of back-firing and pre-ignition, Messrs. Stokes and Cunningham give typical indicator diagrams which are reproduced at figs. 164A and 164B, and they discuss them as follows:

'Glancing over these seriatim, we find that the explosion pressure developed on starting rises to 448 lbs. per sq. in., with a MEP of 78 lbs. per sq. in. on the largest card, but rapidly falls with each successive explosion. The optical diagram, on a different scale, was obtained from a Hopkinson apparatus, which, however, gave no appreciable difference in the MEP outlines to that obtained with the ordinary Dobbie-M'Innes indicator. The next card reveals the degree of steadiness that may be obtained with gas-engine explosions when the conditions are suitable to such.

'Following, we see a normal running diagram obtained any day on average load from one of our large engines. Notice the low explosion effort, 220 lbs. per sq. in., the curve of combustion, and the sustained expansion curve, resulting in a comparatively high MEP of 57·5 lbs. per sq. in.

'Our next diagram introduces a peculiarity of our engine which has been a stumbling-block to many in the past. Before proceeding to discuss it, we would again draw attention to the fact that the two pistons move simultaneously in opposite directions. At the explosion centre the mid crank pin (to which is attached the front piston) is at its inner dead centre, whilst the side crank pins and back piston are at their outer dead centre. When movement commences the front piston travels more quickly than the back piston until approximately half stroke; thereafter the relative rates of motion are reversed. In the usual manner of indicating which we adopt, the indicator diagram shows the pressures acting on the pistons for various positions of the back piston. This diagram is given in the figure by the inner line. When the pressure is, for instance, given by AB, the back piston is at the part of its stroke corresponding to OA, but the position of the front piston corresponds to OC. We can thus draw the indicator diagram

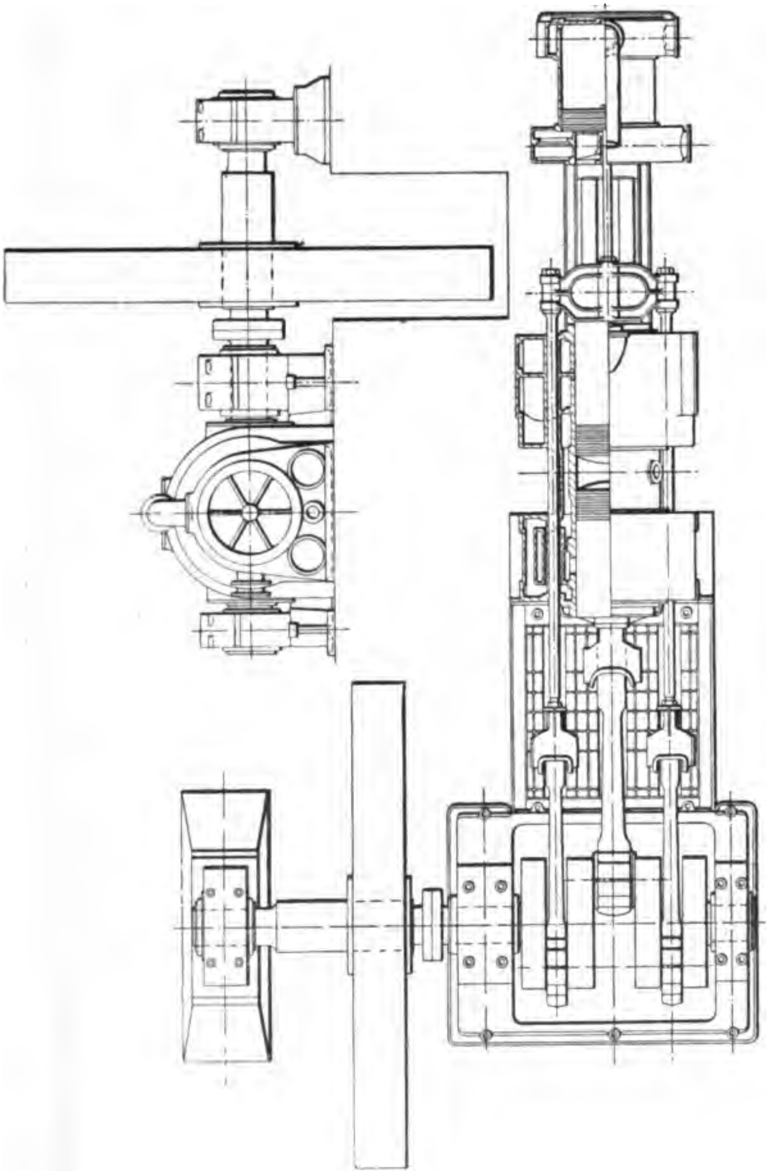
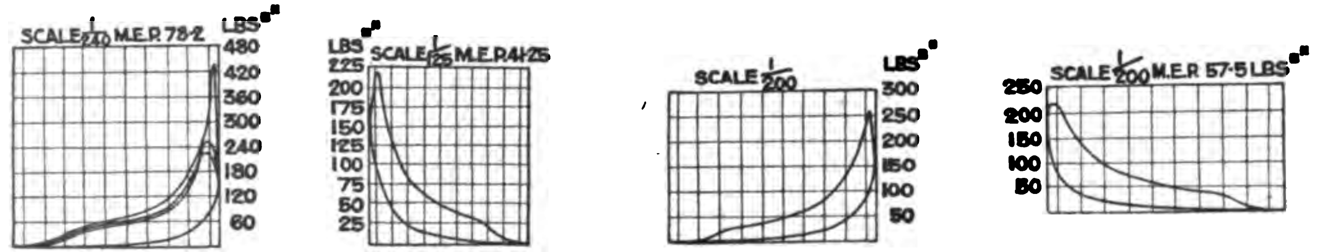
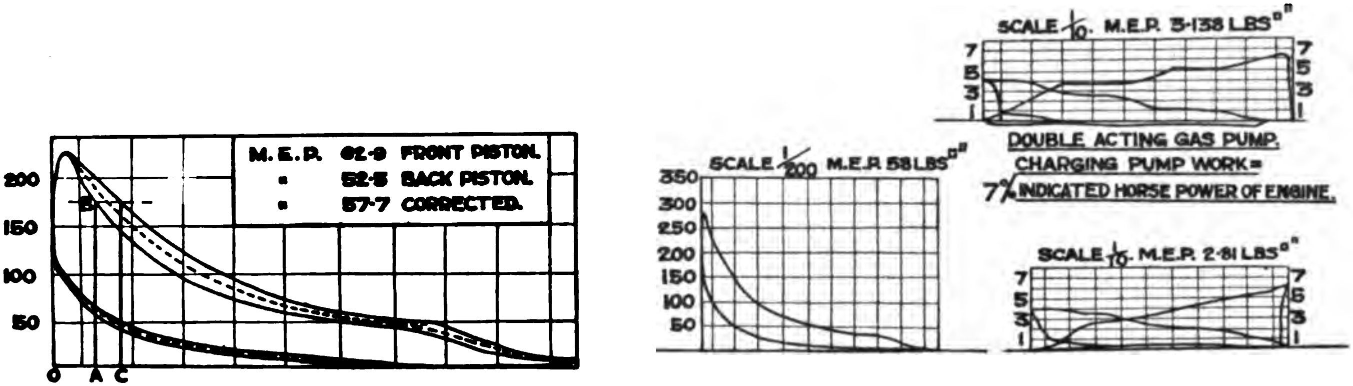


Fig. 163



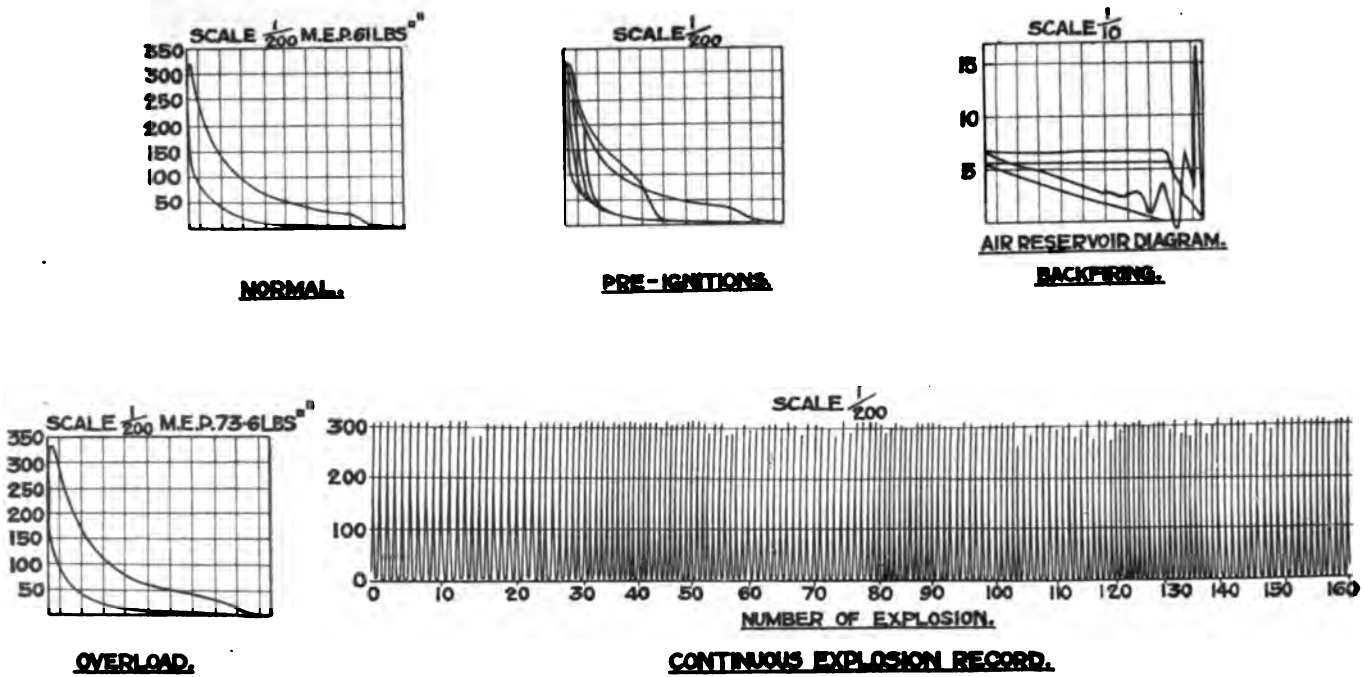
STARTING UP DIAGRAMS.—— OPTICAL INDICATOR DIAGRAM. —— STEADY WORKING, 20 CONSECUTIVE EXPLOSIONS. —— NORMAL RUNNING DIAGRAM.



DOUBLE ACTING GAS PUMP.
CHARGING PUMP WORK = 7% INDICATED HORSE POWER OF ENGINE.

CORRESPONDING DIAGRAMS OBTAINED WITH BACK & FRONT PISTONS.—FULL LOAD DIAGRAM 1020 B.H.P. —— DOUBLE ACTING AIR PUMP.

FIG. 164A



NORMAL.

PRE-IGNITIONS.

BACKFIRING.

OVERLOAD.

CONTINUOUS EXPLOSION RECORD.

FIG. 164B

for the front piston as shown by the outer curve with full line in the figure. Midway between these two diagrams we obtain what is known as the "corrected" indicator diagram, which gives the true MEP for the engine, being the mean of those obtained for the front and back pistons. This corrected diagram is given in the figure by the middle dotted line. The true MEP is about 10 per cent. greater than that obtained from the usual indicator diagram.

'The next diagram is a full load one for a 1000 BHP single cylinder engine driving a dynamo (on water resistance) with the corresponding gas and air pump cards. It is to be noticed that the pump work was only 7 per cent. of that indicated in the working cylinder.

'Below this are examples of pre-ignitions and back-firing obtained a year ago. In the case of back-firing, notice how the normal pressure in the air receiver is rudely disturbed, and the kick back rises to a pressure of 16½ lbs. per sq. in. above atmosphere, accompanied by considerable noise, as previously mentioned. Although somewhat disturbing, no fear need be entertained of breakage on this account, as the receivers are made amply strong to withstand such shocks. It must be clearly understood, however, that such back-firing is of rare occurrence, and immediately on its appearance suitable and simple steps are taken to overcome same. The pre-ignitions shown very clearly indicate their severe pulling-up action on the engine. The normal diagram was obtained on the same engine shortly afterwards. We show also an overload diagram in which the engine was carrying fully 20 per cent. above its normal load. The last diagram is a continuous record of the explosion pressures obtained on a 1000 BHP engine driving a direct current generator at full load.'

They also give heat consumption curves from three tests made respectively at Borsigwerk in Silesia, Herstel in Belgium, and Swanscombe in England.

They are reproduced at fig. 165.

The upper curved lines show the consumption of heat per BHP hour on the scale at the right hand, while the straight lines below refer to the left-hand scale, and show the total heat units per hour required for the BHP shown on the horizontal scale below.

The best brake result is given on the first diagram as about 9300 B.Th.U. per BHP hour at full load, but it would have been more satisfactory if the BHP had been determined in some other way than by assuming it to be given by the blowing cylinder diagram. The second diagram shows a heat consumption of 9700 B.Th.U. per BHP, also at full load, and the third diagram, illustrating the English trials, shows about 10,000 B.Th.U. per BHP at 400 brake horse-power. The tests were made respectively in 1903, 1904, and 1908.

The respective brake thermal efficiencies are, therefore, 27·7 per cent., 26·5 per cent., and 25·7 per cent.

The results are good, but not quite so good as given by four-cycle engines. The highest results which have been claimed for this engine, so far as the authors' knowledge carries them, are found in trials made by Prof. E. Meyer of the Technische Hochschule, Charlottenburg, in August and October 1903. The following particulars are given by Mr. Junge in his interesting book on 'Gas Power' already referred to.

TEST OF A 500 HP BORSIG-OECHELHAUSER GAS ENGINE. (*Meyer*)

Fuel, coke oven gas.
Date of test, October 10, 1903.

Particulars of Engine.

Working cylinder with two pistons	Diameter of cylinder, 26·6 ins. Stroke of front piston, 37·5 ins. Stroke of back piston, 37·3 ins.
Air pump, double acting	Diameter of cylinder, 44·9 ins. Stroke „ „ 19·7 ins. Diameter of front piston rod, 3·45 ins. Diameter of back „ „ 2·78 ins.
Gas pump, single acting	Diameter of cylinder, 23·2 ins. Stroke „ „ 19.7 ins.
Blower	Diameter of cylinder, 65 ins. Stroke „ „ 37·3 ins. Diameter of piston rod, 5·9 ins.

	Number of test	VIII	IX	X	VI	VII
	Time of test	11.40 to 12	12.05 to 12.20	12.20 to 1	10.40 to 10.55	11.05 to 11.2
Revs. per minute (mean)		103·0	107·0	106·1	108·2	107·4
Working cylinder	Mean effective pressure, lbs. per sq. in.	75·0	73·8	69·3	62·3	62·0
	Total IHP	21	839	780	715	707
Blowing cylinder	Total indicated work done equivalent to BHP	616·2	626·6	574·8	488	473·8
Air pump	Mean effective pressure, front	5·09	5·38	5·12	5·56	6·09
	Mean effective pressure, back	3·36	3·58	3·41	3·73	3·94
	IHP consumed	68·3	75·2	71·1	79·0	84·5
Gas pump	Mean effective pressure	3·53	3·50	3·58	3·73	3·84
	IHP consumed	7·7	7·8	7·9	8·5	8·6
Total IHP consumed by charging pumps		76·0	83·1	79·1	87·5	93·2
Mechanical efficiency of blowing engine, per cent.		83·9	84·2	83·3	79·2	78·5
Friction HP consumed on engine		117·3	117·3	115·4	129·2	129·2
Lower calorific value of gas, B.Th.U. per cub. ft.		398·7	393·1	381·9	393·1	396·5
Heat consumption	Per total IHP hour	6587	6547	6508	6666	6627
	Per BHP hour done in blower	8650	8650	8650	9642	9761

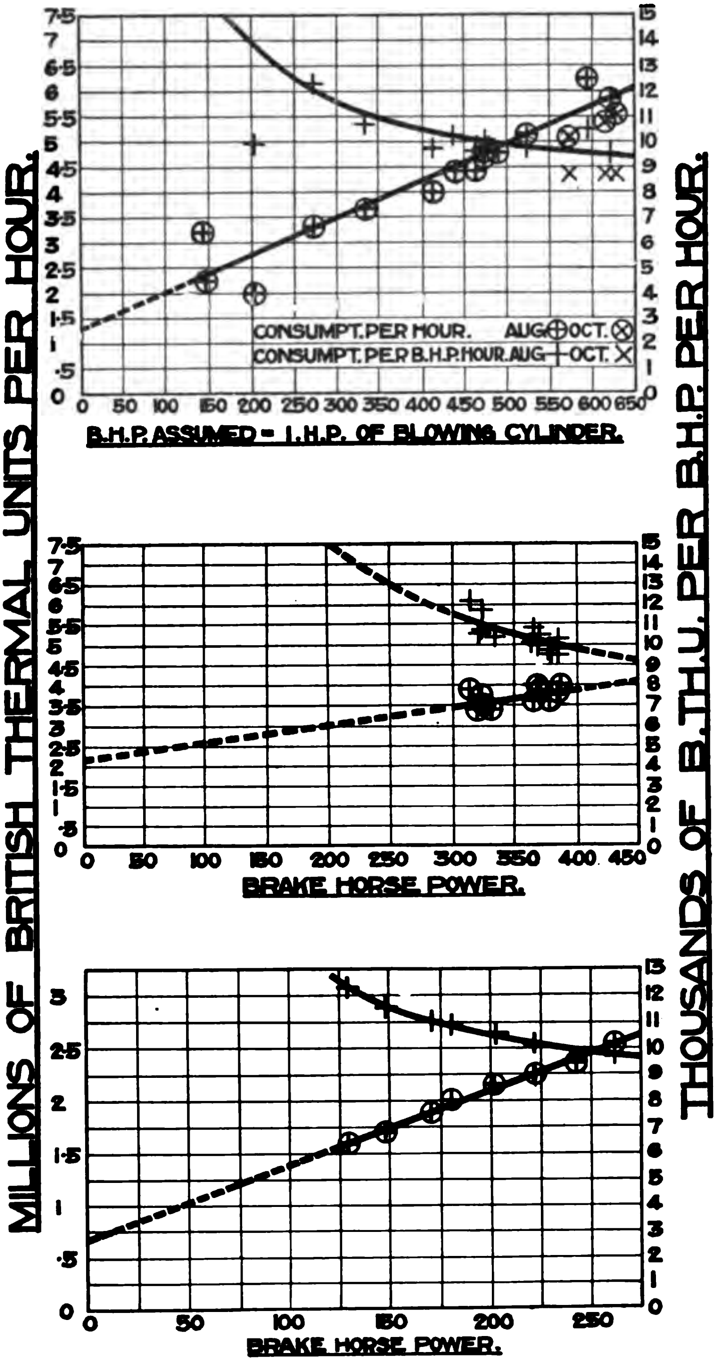

MILLIONS OF BRITISH THERMAL UNITS PER HOUR.
THOUSANDS OF B.TH.U. PER B.H.P. PER HOUR.
CONSUMPT. PER HOUR. AUG ⊕ OCT. ⊗
CONSUMPT. PER B.H.P. HOUR. AUG + OCT. ×
0 ·5 1 1·5 2 2·5 3 3·5 4 4·5 5 5·5 6 6·5 7 7·5
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
0 50 100 150 200 250 300 350 400 450 500 550 600 650
B.H.P. ASSUMED = I.H.P. OF BLOWING CYLINDER.
0 ·5 1 1·5 2 2·5 3 3·5 4 4·5 5 5·5 6 6·5 7 7·5
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
0 50 100 150 200 250 300 350 400 450
BRAKE HORSE POWER.
0 ·5 1 1·5 2 2·5 3
0 1 2 3 4 5 6 7 8 9 10 11 12 13
0 50 100 150 200 250
BRAKE HORSE POWER.

Fig. 165

The following analysis is given to show the variation in the quality of the coke oven gas used in the course of 24 hours:

Analysis of Coke-Oven Gas over 24 hours

Number of test	I.	II.	III.
Time	10 a.m.	4 p.m.	10 a.m.
	Per cent.	Per cent.	Per cent.
H by volume	42·00	48·08	43·8
CO „ „	11·84	10·60	10·2
CH_4 „ „	19·73	18·43	20·3
Heavy hydrocarbons by volume .	2·63	1·18	2·1
N by volume	18·69	15·89	17·9
O „ „	0·20	0·30	0·4
CO_2 „ „	4·91	4·90	5·3
	100·00	100·00	100·0

From these figures it appears that the mechanical efficiency in the English sense varies from about 75 per cent. at the maximum load of 626 BHP to 62 per cent. at 474 BHP. The air and gas pump resistances vary from about 9 per cent. at full load to about 13 per cent. at the lowest load tested calculated on the total IHP.

The thermal efficiency calculated on BHP was 29·7 per cent at the heavy load and 26·3 per cent. at the light load.

The gas consumption was measured by a Pintsch station gas meter, and all precautions were taken to ensure accuracy of indicator reading, as was to be expected from an engineer of Prof. Meyer's reputation and experimental experience.

Nevertheless these brake thermal efficiencies seem to the author to be higher than he expected from an engine of this type.

Lubricating oil used on main cylinder was found to be 1·19 lbs. per hour.

Water used at full load, 5·9 gallons per BHP hour. Water entering at 22° C. and leaving at 42° C.

The following particulars of the recent engines built by Messrs. Beardmore are given by Messrs. Stokes and Cunningham in the paper already referred to.

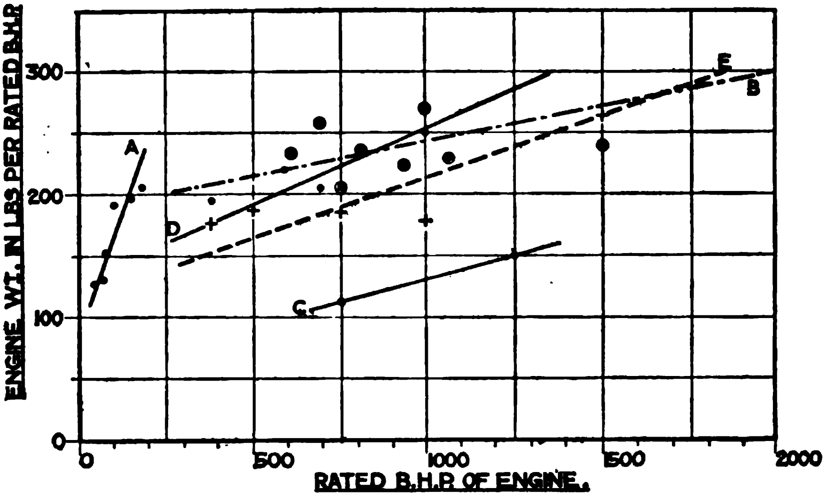
Some Dimensions and Particulars of Oechelhauser Gas Engines. (*Messrs. Wm. Beardmore & Co., Ltd.*) 1909

Brake horse-power	Cylinder diameter	Stroke	Revs. per min.	Piston speed per min.	Approximate weight engine	Approximate weight flywheel
400	24 ins.	30 ins.	130	650	32 tons.	20 tons.
500	26 ins.	30 ins. 37½ ins.	125	625 781	42 tons.	25 tons.
750	30 ins.	37½ ins.	125	781	62 tons.	38 tons.
1000	34 ins.	37½ ins.	125	781	80 tons.	50 tons.
1500	42 ins.	51 ins.	94	797	180 tons.	100 tons.

It is to be noted that in the 500 HP engine two strokes are given; the shorter stroke is that of the back piston; the longer that of the front. In the other cases where only one stroke is given each piston has that stroke.

The piston speeds are the mean piston speed of each of the two pistons, so that to get the effective piston speed, the speeds of the two pistons are to be added.

Messrs. Beardmore guarantee the heat consumption at full load to be between 10;000 and 10,500 B.Th.U. per BHP hour, and the cost of lubrication on the 1000 HP single cylinder is given at 3½*d*. per hour.



A: ◎. Horizontal, single-cylinder, single-acting, four-cycle National engines.
B: ⊙. Large Continental, double-acting, four-cycle tandem engines.
C: ●. Vertical tandem, single-acting, four-cycle National engines.
D: ●. Mather & Platt Koerting, single-cylinder, double-acting, two-cycle engines.
E: +. Beardmore-Oechelhauser, single-cylinder, single-acting, two-cycle engines.

FIG. 166

Messrs. Beardmore have since designed an engine to give 2500 BHP for a single cylinder. It is 48 ins. cylinder diameter, stroke 60 ins.; and speed 80 revolutions per minute. This gives a piston speed of 800 ft. per minute. This large engine must be rated at a higher mean effective pressure than the smaller engines.

Comparison of Koerting and Oechelhauser Engines as to Weight.—The following table has been prepared from the tables of Koerting and Oechelhauser engine particulars given on pp. 239 and 255 to compare the weights of the engines per rated HP with flywheel and without. From this it will be noted that the Oechelhauser is in most powers a somewhat lighter engine per rated BHP than the Koerting when comparison is made by the net weights of the engines without flywheels, but when flywheels are included the reverse is the case.

This is due to the fact that the Koerting engine, having two impulses per revolution against one of the Oechelhauser, only requires flywheel weight which ranges from 56 to 72 lbs. per BHP, while the Oechelhauser flywheel weights range from 112 to 149 lbs. per BHP. It is remarkable, however, to find a single-acting engine somewhat lighter than a double-acting. This appears to be due to lightness of the Oechelhauser engine beds permitted by the transmission of most of the working stresses to the crankshaft by the connecting-rods, side rods, and crossheads. The absence of packing glands on the Oechelhauser also tends to lightness.

WEIGHTS OF KOERTING AND OECHELHAUSER ENGINES

Rated BHP.		Cylinder dia.	Revs. per min.	Piston speed per min.	Approx. net wt. of engine.	Wt. of fly-wheel.	Wt. of engine per BHP.	Wt. of flywheel per BHP.	Wt. of engine and fly-wheel per BHP	ηp in lbs. per sq. in.
		ins.			tons	tons	lbs.	lbs.	lbs.	
Mather & Platt Koerting single double - acting power cylinder.	400	—	110	—	35	10	196	56	252	—
	500	—	110	—	37	13	166	58	224	—
	600	—	100	—	59	16	220	60	280	—
	700	—	95	—	65	20	208	64	272	—
	1000	—	80	—	103	32	231	71·6	303	—
Beardmore Oechelhauser single cylinder, single-acting power cylinder; 2 pistons.	400	24	130	650	32	20	179	112	291	45
	500	26	125	625 781	42	25	188	112	300	44·5
	750	30	125	781	62	38	185	113·5	298	44·8
	1000	34	125	781	80	50	179	112	291	46·7
	1500	42	94	797	180	100	269	149·3	418	47·0
	2500	48	80	800	—	—	—	—	—	57·0

Fig. 166 shows the weights per BHP of these two engines plotted against the BHP, together with the similar weights of (A) single-cylinder, single-acting, four-cycle engines, (B) tandem double-acting, four-cycle engines, and (C) vertical tandem single-acting, four-cycle engines. It will be seen that although an increase in weight per BHP (with increased power) is apparent in both Koerting and Oechelhauser, yet 'the experimental points are not regular. Some irregularity in design has prevented the true increase to be readily seen. The increase is assumed to be linear, and the line D shows the Koerting while E gives the Oechelhauser.

The Koerting engine is lighter than the single-acting, four-cycle engines, but slightly heavier than the others.

The Oechelhauser follows very closely the weight of the horizontal tandem double-acting engines. The comparison of weight is interesting, and it is somewhat surprising to find the four-cycle engines relatively so light.

CHAPTER III

IGNITING ARRANGEMENTS OF GAS, PETROL, AND OIL ENGINES

SECTION I

GENERAL

HOWEVER perfect the theoretic cycle of an engine, however admirable its construction, in the absence of an effective igniting device all the skill and energy expended are without avail. The engine is a useless mass of metal requiring power to set it in motion instead of itself furnishing a source of power for driving other machines.

In the earlier stage of gas-engine manufacture the ignition methods adopted provided a most fruitful source of trouble; the problem of ignition has proved by no means simple, and the time and trouble expended on its solution would not be suspected from an examination of the igniting gear of any good modern engine.

In the non-compression engines the problem is comparatively simple—to inflame a volume of explosive mixture enclosed in a cylinder, so that the explosion is confined within the cylinder, and no communication is open to atmosphere. This is to be repeated regularly and with certainty at rates varying from 60 to 150 times per minute, depending upon the speed of the engine. In the earliest designs what may be called the touch-hole method naturally suggested itself ; the piston after taking in its charge crossed a small hole and sucked a flame through it into the cylinder, the hole being either so small as to occasion no substantial loss of pressure upon explosion, or covered by a small valve closed by the interior pressure. This was the earliest flame method. Then came the idea of using the electric spark and so completely closing up the cylinder, and later on a return to flame ignition, using a double flame, viz. an outer one to produce an intermediate one, this latter being carried by a pocket, or hollow cock, to the mixture in the engine cylinder. Then the idea of spongy platinum suggested by the well-known Doebereiner's hydrogen lamp. After this the heating of metal tubes or masses and ignition of the mixture by contact with the heated surface. Then electrical ignition again, this time by heating a platinum wire to incandescence.

More recently electrical ignition, using specially designed 'low-tension magneto' machines, has been very extensively adopted for stationary gas engines of all sizes, with quite satisfactory results in general everyday practice; while finally it appears possible that high-tension magneto ignition, already almost universal in automobiles, may in the near future be largely employed also in the heavier classes of work.

A noteworthy development of the coil-and-battery system recently introduced is the well-known Lodge method of ignition; this is much employed in large gas engines; a detailed account is given later in this Chapter.

Ignition methods may, then, be classed in four groups:

(1) Flame methods;
(2) Incandescence methods;
(3) Methods depending on 'catalytic' or chemical action;
(4) Electrical methods;

and an account will next be given of the more important of the many devices that have at various times been proposed as solutions of the problem of ignition.

1. Flame Methods

The earliest really efficient igniting valve is that described by Barnett in his specification of 1838. It is the parent form of that once extensively used valve, viz. the 'Otto,' now displaced by tube and electrical ignitions.

Barnett's Igniting Valve.—Fig. 167 shows a vertical section and a plan of this valve. It consists of a conical stopcock with a hollow plug; the shell contains two ports, 1 and 2, of which 1 opens to the atmosphere while 2 communicates with the cylinder. The plug of the cock has one port, 3, so arranged that it may open to the atmosphere port or to the cylinder port in the shell, but cover enough is left to prevent it opening to both at the same time. In turning round it closes on the atmosphere before opening to the cylinder.

A gas jet burns at the bottom of the shell, and within the hollow of the plug, the ports 3 and 1 being long enough and wide enough to allow the air free circulation as shown by the arrows. The flame must not be too large or it will fill the whole interior with gas and prevent air getting in; in this case the gas burns at the port 1 in the air and will not enter the cock. Suppose it to be burning regularly in the cock as shown in the drawing, then if the plug be suddenly turned round so that port 3 closes upon the atmosphere port 1, and opens to the cylinder port 2, the air supply will be sufficient to keep the flame living till the mixture contained in 2 reaches it. The explosion then occurs. The port 2 is of the same shape as 1, so that the flame causes the gases to circulate the same as the air did when open to it; the mixture comes in contact with the flame by circulating through the plug. If the port 2 is made so small that no circulation occurs, the ignition becomes uncertain, as the gases must then reach the flame by diffusion, which is a slow process, and the flame may be extinguished before they arrive at it. The explosion, of course, extinguishes the

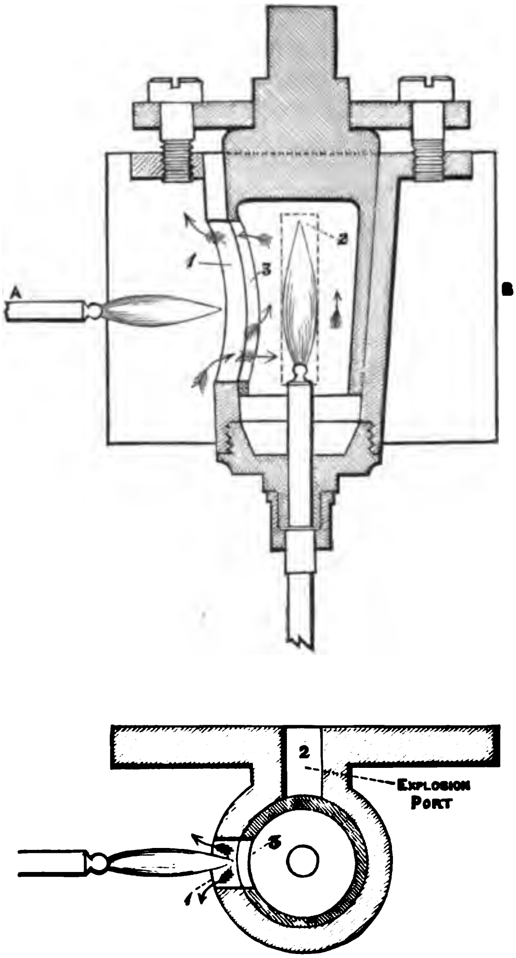


Fig. 167.—Barnett's Igniting Valve (flame)

flame, but when the plug again rotates so as to open to the air, the external flame relights it, and it is ready for the next ignition.

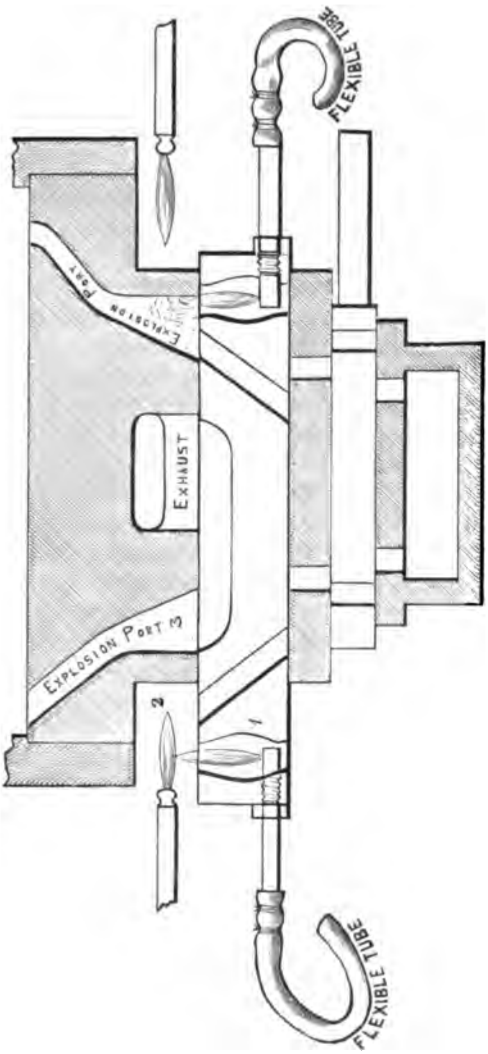


Fig. 168.—Hugon Flame Igniting Valve.

Hugon's Igniting Valve.—In the small Hugon engine Barnett's method was first applied in a fairly successful manner.

The valve is shown in section at fig. 168.

The sectional plan, fig. 168, shows the internal flame lit and burning in the ignition port 1; the external flame 2 burns close to it in this position, so as to be ready to light it when wanted. The gas for the internal flame is supplied under higher pressure than that of the ordinary gas main by a bellows pump and small reservoir through a flexible rubber tube. For the external flame the gas is used at the ordinary pressure.

When ignition is required, the valve moves rapidly forward, causing the port 1 to close to atmosphere first, and then to open to the cylinder port, as shown at the other end of the slide.

The explosive mixture which fills the port is at once ignited, and the flame finds its way from the port into the cylinder itself; the port is necessarily filled with pure explosive mixture free from any admixture with exhaust gases as all the mixture before entering the cylinder must pass through it, and so sweep before it any burned gases into the cylinder. Hence the mixture in the port will be more ignitible than that in the cylinder, as the mixture there is diluted in part with exhaust gases while that in the port is free from them.

The explosion is thus exceedingly certain and regular; when it occurs it extinguishes the internal flame, and at the same time its superior pressure forces back the gas in the rubber tube while the port 1 remains open to the cylinder.

The return of the slide again opens it to the atmosphere, and here is seen the necessity of using the gas under some pressure. Before it can relight at the external flame, the products of combustion must be expelled from the gas pipe; if the gas were under only the ordinary gas main pressure, there would be no time for this, and the valve would return to ignite without a flame. The expedient of increasing pressure is somewhat clumsy, but it acts fairly well. The port 1 is made large to give space for the air necessary to support the flame while the ignition port is passing from atmosphere to cylinder port. At the moment of explosion the cylinder is completely closed from the air.

The explosion is therefore completely contained within the cylinder, and no sound is heard.

In an engine at the Patent Office Museum Mr. S. Ford considerably improved the igniting arrangement by intercepting the rush back to the gas pipe by a light check valve; he was thus able to use gas under the ordinary gas main pressure and dispense entirely with Hugon's gas pump and reservoir. The explosion, instead of forcing a considerable volume of burned gases down the gas pipe, simply closed the check valve, which opened as soon as the igniting port reached the air again, and so gave the gas stream at once.

Otto's Igniting Valve.—The igniting valve used in the Otto and Langen engines is a further development of Barnett and Hugon's igniting devices.

As applied to the compression engine there is, however, one alteration, very slight, but very essential.

In the Lenoir and Hugon engines, as in the Otto and Langen, the pressure in the cylinder is the same, or in some cases less, than that of the external atmosphere, that is, before ignition. It is therefore an easy matter to transfer a flame burning quietly in the air to the cylinder without danger of extinction. When the gases to which the flame is to be transferred exist at a pressure some 40 to 50 lbs. per sq. in. superior to that of the flame itself, it is not so easily seen how the flame is to be transferred without extinction. Generally described, the arrangement is as follows. A small quantity of coal gas is introduced into the upper part of a cavity in the ignition slide; being lighter than air, it remains separate from it, and has no tendency to mix with the air beneath it, except by the slow process of gaseous diffusion. At the surface of contact with the air it is ignited, and burns with a blue flickering flame. The movement of the slide cuts off communication with the outer atmosphere, and very shortly thereafter opens on the admission port of the engine; but before doing this it opens to a small hole communicating with the cylinder. This hole communicates with the gas passage in the upper part of the slide, so that the gases under pressure enter and force the gas downwards, the pressure rising in the port more slowly than would occur if the main port opened at once. The pressure is therefore nearly level with that in the cylinder when the main port opens, and the flame still burning at the surface of junction between the gas and air ignites the mixture. If the pressure were not raised in the igniting port by pressing the gas downwards, and thereby avoiding a rush past the flame portion, the rush would often extinguish the flame and an ignition would be missed. The apparent difficulty of transferring the flame from atmosphere to 40 lbs. above it is thus simply and beautifully overcome. By using a portion of gas in the upper part of the valve cavity, the difficulty of the blow back of explosion down the gas supply pipe is also overcome, as the gas supply can be cut off before the explosion or compression pressure comes on. It is cut off just before the valve closes the flame port to atmosphere.

Fig. 169 is a vertical section showing the flame cavity in the slide, in the act of introducing coal gas at the upper part and inflaming it at the surface of junction between gas and air.

The slide A contains a forked passage, B, communicating by the lower branch with the air inlet C, and by the upper with the funnel F, which are both in the cover D, which holds the valve against the engine face. The jet G has a flame constantly burning into the funnel, which becomes heated, with the effect of drawing a current of air through the forked passage when its ports are in proper position; the direction of the current is shown by the arrows. The pipe J supplies coal gas, which passes along the gutter I cut in the cover and valve faces into the forked passage C, and thence to the funnel F, where it is inflamed and burns as shown. When the movement of the slide cuts off communication with the atmosphere it also closes the gutter I and terminates the supply of coal gas from the pipe J, but the upper part of the forked passage contains gas; a flame therefore flickers as shown. Just before B opens on the port L, fig. 170, the hole K, fig. 169, opens and the pressure from the explosion space causes a flow into B, forcing before it the gas contained in the hole, thereby intensifying the flame by making the gas pass more into the air and bringing about the equilibrium of the pressures. When B opens on L the flame is vigorous and at once fires the whole charge in the explosion chamber. Fig. 170 shows the slide with the port B about to open on L. Fig. 171 is an end elevation of the valve and cover, showing the ports and gutters dotted and lettered in the same position as in fig. 170. The method is carried out completely and is very perfect; it is somewhat slow in action, as it depends on a proper ventilation of the forked passage and the complete replacement of the burned products by fresh air before the gas can burn properly in the cavity. If the engine be run more rapidly than the draught of the funnel can clear out the passage from the burned gases, then the flame cannot be lit in it and an ignition will be missed.

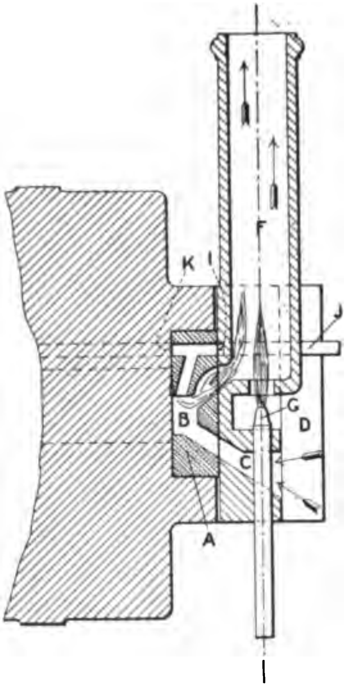


FIG. 169

It is a method exceedingly successful when ignition is not required too frequently, but very troublesome and uncertain for rapid ignition. The Otto and Langen engine only made 30 ignitions per minute, and the Otto compression engine makes but 80 ignitions per minute at full power; its efficiency is good at these rates, but at 150 per minute it is too slow in action.

Clerk's Igniting Valve.—The method of igniting the charge used by the author was quite different from the other flame methods already described; the difference was necessitated by the greater rapidity of ignition in engines with an impulse in every revolution.

To ventilate the igniting port in the Otto and Hugon slides requires time, which cannot be given when the frequency of the ignition approaches 150 to 200 per minute.

To meet this difficulty the author invented several methods, both flame and incandescence, and the one to be described is very reliable and rapid, as many as 300 ignitions per minute having been made with it experimentally, or at the rate of 5 ignitions per second.

A portion of the explosive charge is allowed to pass from the motor cylinder through a regulated passage to a grating placed at the end of a cavity in the slide, and is there ignited by a Bunsen flame; the grating prevents the passage back of the flame, and the mixture burns in the cavity without requiring the presence of the external atmosphere. At each end of the cavity there is a port opening to opposite sides of the valve, the one for lighting the gases streaming from the grating, the other for communicating with the interior of the cylinder at the proper time. The communication with the cylinder is not made until the outer port cuts off from atmosphere, and the flow of the gases is so regulated that while this is being done, the flame still continues to be fed by fresh supplies. It is evident that if too great a current be sent in, the

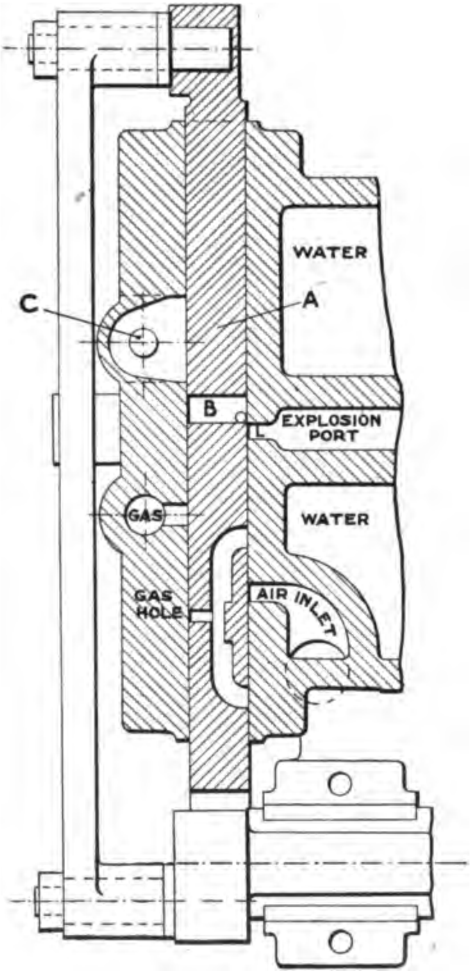


FIG. 170

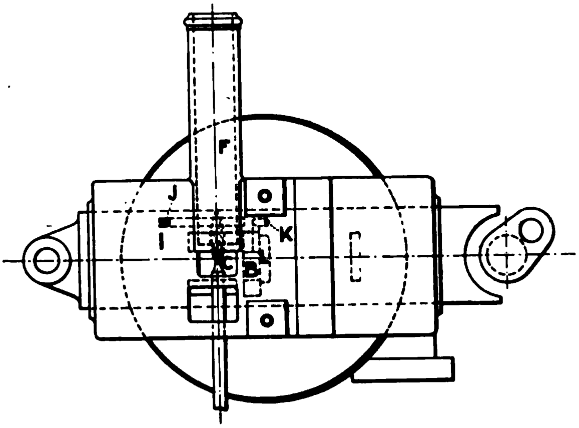
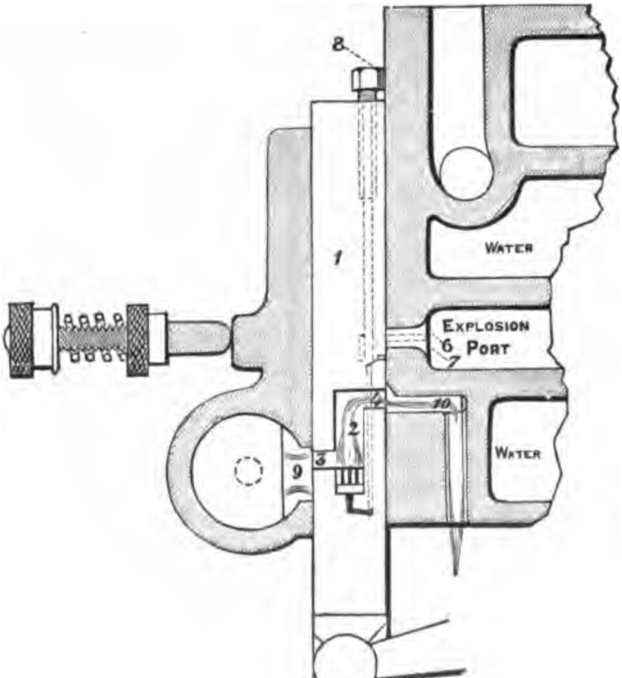


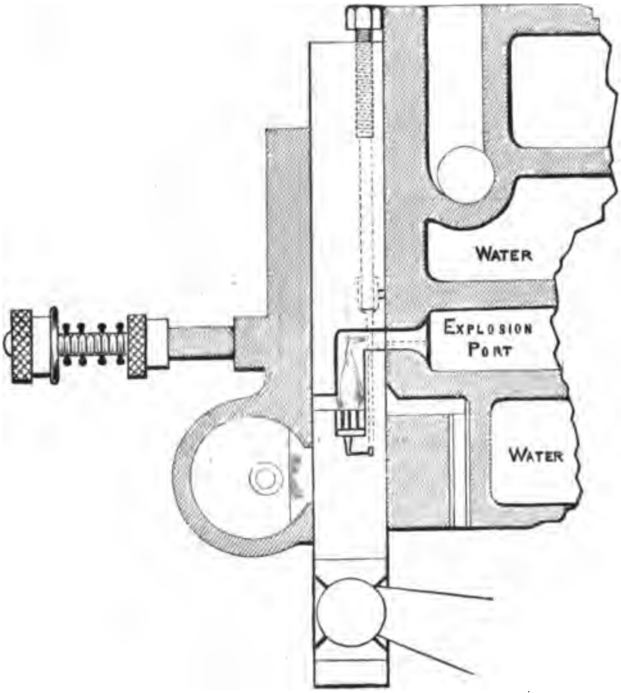
FIG. 171



Valve in position of flame lighting at external flame.

FIG. 172.—Sectional Plan, Clerk Igniting Valve.

pressure will soon become equal to that in the cylinder, and then the flow towards the cavity will cease and the flame become extinguished; this is guarded against by properly proportioning the flow by the check pin. The pressure in the cavity when its port opens on the cylinder port is still slightly less than that in the cylinder, and the gases from the cylinder accordingly enter and are ignited. By using gas and air already mixed in proper proportion, the necessity of ventilating is removed, and it becomes possible to ignite at the rate required by the system of impulse in every revolution. Without this it would be almost impossible to get a passage cleared out in time to allow of so frequent ignition, by a coal gas flame burning simply in air. It was first used by the author in an engine working in February 1878, and has subsequently been used by Wittig and Hees, and by Robinson, in the Tangye engine. In the form here described it was first used in November 1880.



Internal flame exploding mixture.

FIG. 173.—Sectional Plan, Clerk Igniting Valve

Fig. 172 is a sectional plan of the igniting slide and cover as well as the passage into the combustion space. The valve 1 contains the cavity 2, furnished at the ends with the ports 3 and 4; at the end 3 is placed a grating communicating behind with the explosion port 6 by a small hole, 7, and a gutter in the valve face. A long pin, 8, screwed into the end of the slide controls the gases entering the space behind the grating, and if need be can cut off communication altogether. When the valve is in the position shown in the drawing, the mixture is beginning to flow through the grating into the space 2, and is ignited by the Bunsen flame 9 lying up against the valve face. The Bunsen flame lies so close to the grating that immediately inflammable mixture comes it is lighted before it can get time to fill the cavity; if allowed to accumulate in the cavity before lighting, a slight explosion ensues and a disagreeable report is produced. The flame at the grating burns in the cavity, discharging into the passage 10, and from thence to the atmosphere. The movement of the slide cuts off communication with the atmosphere, first on the Bunsen flame side, and then on the inside of the valve; very shortly after, the port 4 opens on the port 6 leading to the cylinder, and the gases then taking fire communicate the flame to the whole contents of the compression space. In fig. 173 the flame port in the valve is full open on the explosion port of the engine.

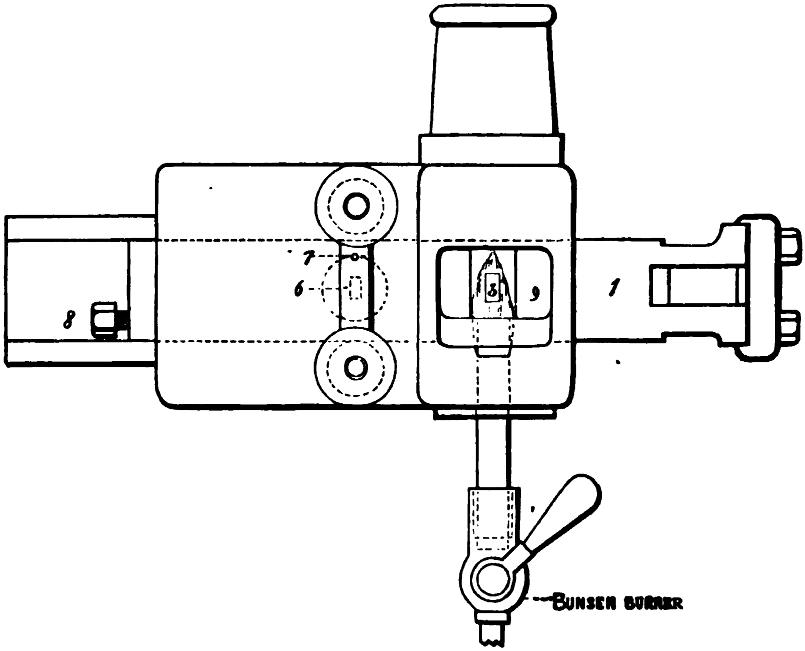


FIG. 174.—End Elevation, Clerk Igniting Valve

The slide then moves past the port and back to the first position, where the operations described are repeated and ignition again occurs.

This arrangement is very rapid in action, and is capable of igniting with the utmost regularity at a rate so high as 300 times per minute, which is far in excess of the requirements of the engine. Fig. 174 shows the Bunsen flame burning against the face of the valve, ready to ignite the gaseous mixture.

Brayton's Flame Ignition.—The Brayton method of ignition has already been described shortly in the description of the engine in the first volume. It is so beautiful and instructive that it merits further discussion.

The action will be made clearer by describing a well-known laboratory experiment (fig. 175).

A piece of wire gauze, *a*, held a few inches from the Bunsen lamp *b*, the gas being turned on, will prevent the flame when lit above it from passing back through the gauze to the burner. The gauze may be moved through a considerable distance from the Bunsen tube without extinguishing the flame. The mixture of gas and air streaming from the Bunsen passes through the gauze, and, although ignited above, the heat is so rapidly conducted away by the gauze, that the flame cannot pass through its interstices to the lower side. If an explosive mixture be confined under, say, 30 lbs. per sq. in. pressure in a vessel, and a pipe from it (fig. 176) leads to a pair of perforated plates with gauze between them, then the cock *b*, being opened gently (the valve *c* being previously open), the mixture will stream through the plates into the atmosphere, and, if ignited, will burn without passing back. If the cock *b* is opened suddenly, a greater rush of flame will occur, diminishing again if it is partly closed.

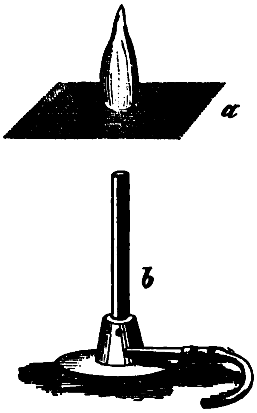


FIG. 175.—Bunsen Flame burning above Gauze

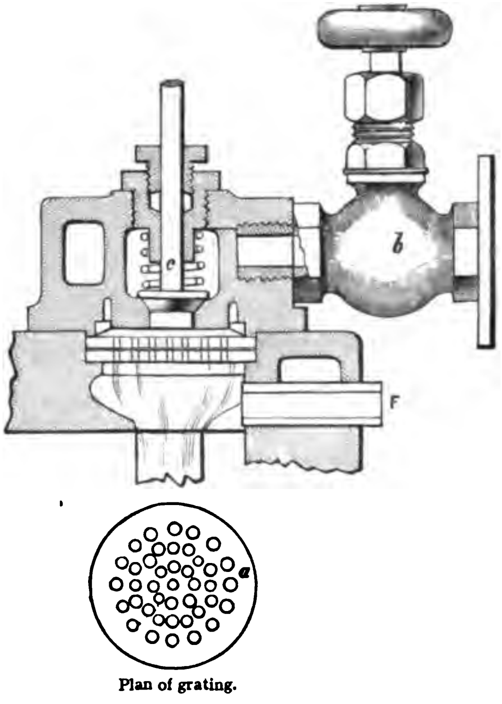
So long as enough mixture passes to preserve alive the flame, then any increased quantity passing from the reservoir will be burned, the little flame increasing or diminishing as the opening of the stopcock valve is increased or diminished.

The action of the ignition in the Brayton engine is exactly similar. The pressure on the flame side of the grating is slightly below that existing on the other side ; the stream of cold gases entering the engine cylinder immediately becomes flame on the grating, and so expands, the volume of flame being changed as required by the valve action of the engine.

This method was successfully carried out in the Brayton engine. The lack of economy was not due to the ignition, but to the use of it under unsuitable circumstances. It is quite possible that this system in a better combination may come into use in large gas engines of the future.

2. INCANDESCENCE METHODS

The ignition of explosive mixtures by contact with heated metallic surfaces was first proposed by the late Sir. C. W. Siemens, and after him by the American, Dr. Drake. Dr. Siemens, in one of his gas-engine patents, proposes to ignite the mixture by passing it through an iron tube, which is heated to redness by a flame outside it.



Plan of grating.

FIG. 176.—Brayton Grating and Valve

Dr. Drake constructed an engine in which the ignition was effected in a similar manner. The difficulty is found in the rapid oxidation of the tube, and the consequent necessity for frequent renewal. Numerous attempts have also been made to heat a portion of the interior surface of the cylinder, so that at a suitable time the mixture might be exposed to it and fired.

The first arrangement of incandescent ignition successfully applied to a compression engine is the invention of the author, and is described in his patent No. 3045 of 1878. It was used in an engine exhibited at the Royal Agricultural Society's Show, Kilburn, in July 1879.

Clerk's Igniting Valve.—Fig. 177 is a sectional plan of this valve in position. Fig. 178 is a separate view of the valve looking upon the face, and fig. 179 is the platinum cage, full size, taken out of the valve.

The platinum cage consists of a box of platinum plate, with numerous platinum ribs running across it. They are secured by rivets running completely through, small platinum washers serving to keep the plates at equal distances apart. The valve receives this cage in a cavity, and it is tightly packed in its place with asbestos and slate packing, a covering plate screwed down upon it securing the whole in position. To start the engine, the reservoir containing gas and air under pressure is opened; the small tap 1 permits mixture to flow through the diaphragm 2 (made like the Brayton grating), and the mixture is ignited at the small door below, which is then closed. The flame flows through the platinum cage, heating up its plates to a white heat in a few seconds. On opening the starting cock of the engine, it moves, and brings the igniting port 4 on the cylinder port 5, at the same time opening on the port 6 in the cover, leading into the cavity 7. The mixture in the cylinder then rushes through the cage, becoming ignited, and the explosion reaches the cylinder; the cavity 7 is so proportioned that each ignition sends a measured

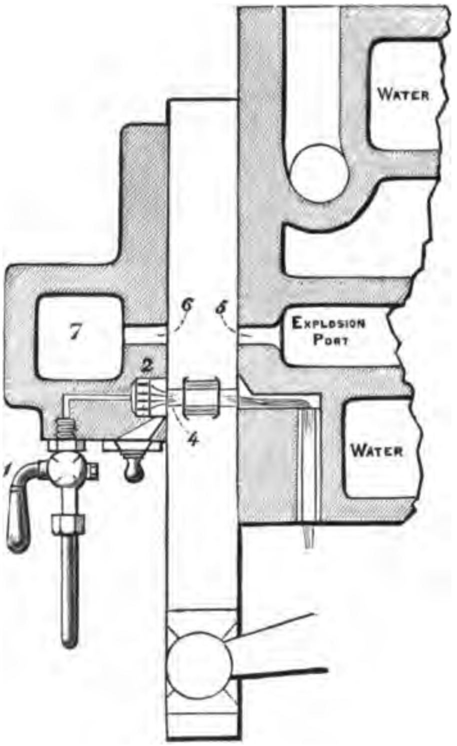


Fig. 177.—Sectional plan, Clerk indandescent platinum igniting valve

quantity of flame through the cage into it; the heat of the explosion at every turn therefore supplies heat to the platinum. This added heat is sufficient to keep it white hot. So long as the engine is supplied with gas it gets an ignition at every revolution, and a portion of that heat goes to the platinum to make up for loss by conduction. The heating flame used in starting the engine is dispensed with immediately after starting, and the engine runs continuously without outside flame. This method is exceedingly reliable and rapid, but is not suited for the governing arrangements of small engines.

Tube Ignition.—The relatively simple, inexpensive, and reliable hot-tube method of ignition has been very largely employed, and tubes of wrought iron, porcelain, nickel, nickel steel, or platinum are used for the purpose. Wrought iron rapidly oxidises, but is cheap and easy to replace; an example of its use is given below. Porcelain is rapidly heated, resists the chemical action of the flame well, and is cheap; it is, however, brittle, and cracks at once on contact with water. Nickel withstands the flame action very well, and in ordinary practice porcelain and nickel tubes came to be very generally used. Platinum has proved to be the most satisfactory material, but its excessive costliness rendered its use prohibitive in all but the small engines of, e.g., the earliest automobiles and motor boats.

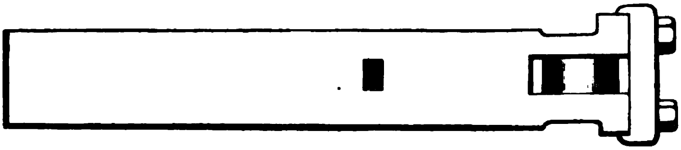


FIG. 178.—Face of Valve with Platinum Cage

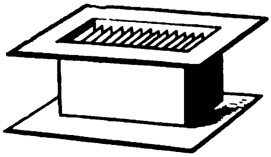


FIG. 179.—Platinum Cage

Experience has shown that to obtain regular and satisfactory ignition the hot tube should not exceed certain proportions; if too great in diameter eddying motions are set up in the gas within it and the ignition becomes irregular. The entering mixture should form an advancing 'plug' filling the bore of the tube, in order to obtain the best results; vertical tubes have been found preferable to horizontal in many cases. The range of sizes usual in practice is from about $\frac{3}{16}''$ bore × 1″ long to ½″ bore × 3½″ long.

The hot tube has now been superseded in all but a few cases of small gas and oil engines by electric methods as described later.

Fig. 180 exhibits an adaptation of Siemens' arrangement by Mr. Atkinson in the 'differential' engine which was shown at the Inventions Exhibition in 1885. The wrought-iron tube 1 is heated by the Bunsen flame 2, the non-conducting casing 3 preventing loss of heat. At the proper instant the piston uncovers the hole 4 into which the tube is screwed, and the mixture entering under pressure becomes ignited on reaching the heated zone. This arrangement was very simple and worked well; care had to be exercised to avoid over-heating the tube, so as to avoid rupture by the explosion. The wrought-iron tube was inexpensive and readily replaced when oxidised or cracked.

Fig. 181 shows in section the hot tube and starting device of Messrs. Andrews' Stockport-Otto engine.

The timing valve F opens into the port above the admission valve and is operated by the cam-actuated lever D. The incandescent metal igniter tube G, heated by an external Bunsen flame H in the usual manner, has a smaller internal tube passing into it as shown, from the space controlled by the timing valve; an annular space remains between this inner tube and the igniter tube, and provides communication with the space closed by the valve A, used for starting purposes only.

At the proper instant the timing valve is opened by the lever D, admitting compressed fresh mixture from the port above the admission valve to the tube G, by way of the internal tube. On reaching the hot zone the mixture immediately ignites and the explosion is communicated to the cylinder contents *viâ* the port. The chamber closed by the valve A serves to ensure a sufficient rush of gas through the tube G to make it certain that fresh explosive mixture shall reach the heated zone; the timing valve F remains opened long enough to allow

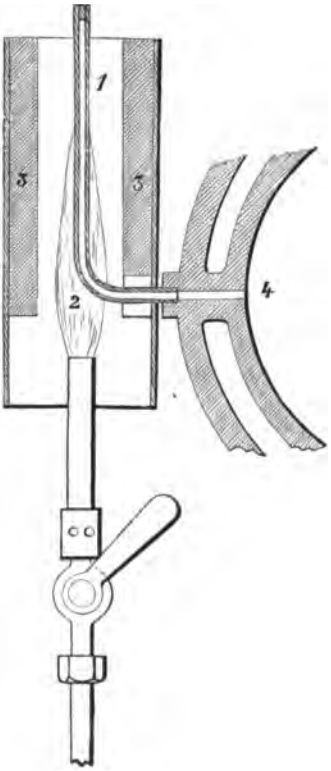


FIG. 180.—Hot Tube Igniter

the contents of the igniting tube and all communicating spaces to be discharged through the exhaust valve in readiness for the next cycle.

Fig. 182 illustrates an arrangement used by Messrs. Crossley in their 9 HP normal 'Otto' gas engine. A portion of the compressed charge was admitted from the compression space to the hot tube R, maintained at a red heat in the usual manner; the passage of the explosive mixture to the igniter was regulated by a timing valve O, operated by the lever Q and cam P. The valve O was double seated, and during the compression stroke the valve face nearest the cylinder was held to its seat by a strong spring; the hot tube was open to the atmosphere, and thus remained empty until the moment required for ignition.

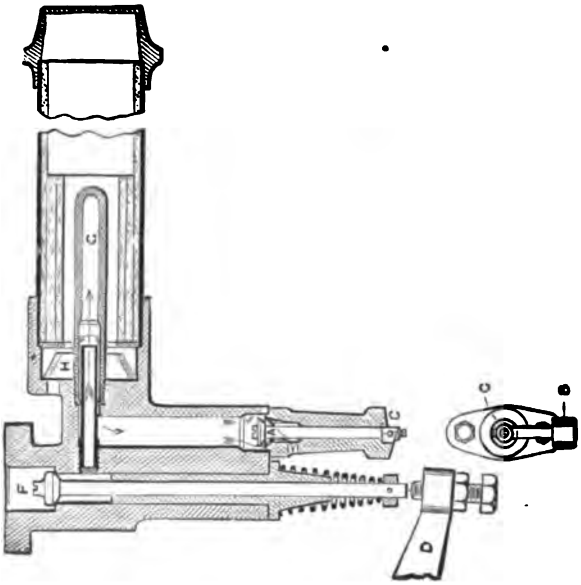


FIG. 181.—Stockport Otto Engine (section incandescent tube and starter)

As the timing valve passed over to its other seat, a very small portion of the compressed mixture was discharged into the air; this served to clear out the burned gases and allow pure explosive mixture only to reach the hot zone of the igniting tube when the outer valve became seated; in this way the ignition was satisfactorily effected.

Fig. 183 illustrates very clearly the hot tube igniting arrangement used by the National Gas Engine Company.

The tube C is of porcelain mounted in an asbestos-lined metal casing, and heated by a Bunsen ring flame burner using about 2½ cub. ft. of town gas per hour. The timing valve B, operated by the lever A

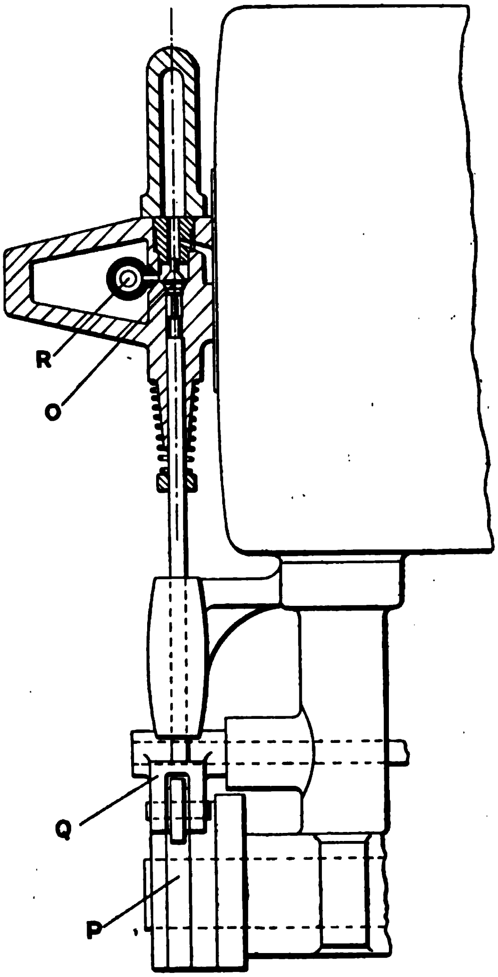


Fig. 182

and cam, admits fresh explosive mixture from the cylinder by way of the passage E at the proper moment; on reaching the hot zone of the tube, ignition occurs and the explosion follows. When the exhaust valve opens, the burnt gases in the porcelain tube and space G are discharged through the passage E. To effectively clear the passage E, and ensure that fresh mixture shall reach the hot tube at the next firing instant, the 'vent tube' F is provided; any burned gas remaining

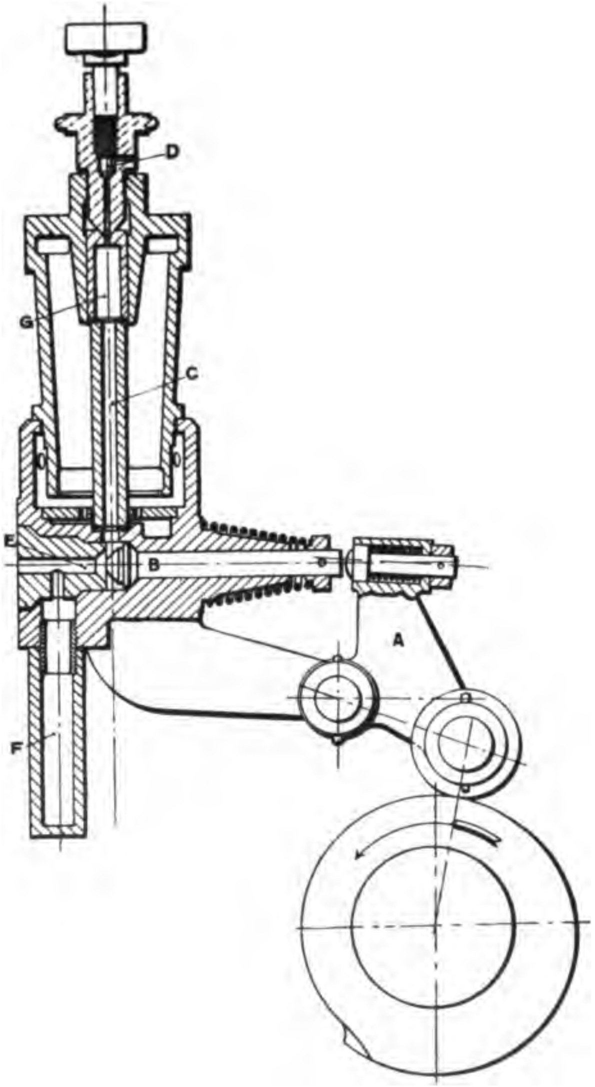


FIG. 183

in the passage E is, during the next compression stroke, packed into F, so that when the timing valve B opens fresh mixture only shall pass on to the igniting tube. Above the igniting tube a packing space, G, is also provided into which any burned gas remaining in the tube from the previous explosion is compressed by the incoming fresh charge; a sharp and regular ignition is thus obtained. When starting the engine, half-compression cams are used, and in order to get then an ignitible mixture into the porcelain tube the relief valve D is opened; this permits a fine stream of combustible mixture to flow from the cylinder, through E, the tube, and G, into the atmosphere; immediately firing commences the valve D is closed.

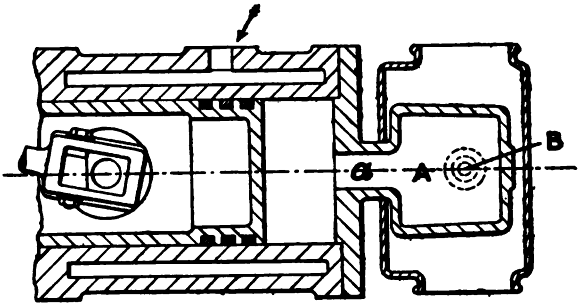


FIG. 184

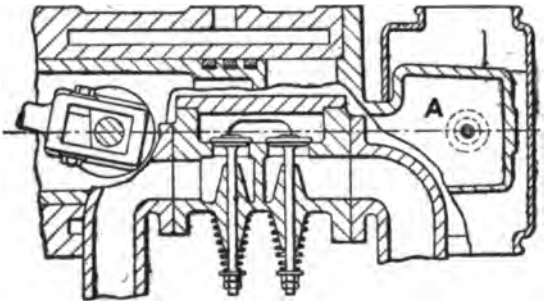


FIG. 185

Ignition by tube gave quite satisfactory results when well designed, especially in engines of up to 20 or 25 HP using town gas. With engines using poor gas it was not so suitable; in cases where suction gas was used there was frequently no supply of gas under pressure available for the Bunsen burner.

During a long period tube ignition was employed without a timing valve, ignition occurring when the compressed fresh mixture reached the hot zone of the tube, the precise instant of firing being determined by moving the heated zone towards or from the cylinder.

Guldner and others have pointed out that this simple arrangement is to some extent automatic in action, as ignition of the entire charge will only take place when the mixture enters the hot tube at a lower velocity than that of the firing flame outwards, and hence has a tendency to occur at or near the dead centre position of the crank pin.

With engines of more than 20 to 25 HP the danger of pre-ignition, and the necessity of exactly and positively controlling the instant of firing, rendered timing valves necessary. The timing valve being exposed to a constant rush of flame, has given a good deal of trouble, and required frequent renewal; moreover the combination of timing valve and hot tube necessitates the occurrence of the ignition at some little distance from the cylinder, and with large cylinders especially this is objectionable; the more modern electrical methods permit ignition to occur at two or more points simultaneously, actually within the combustion chamber.

Open tube ignition, i.e. hot-tube ignition without timing valve and where the heated tube is constantly in communication with the combustion chamber, was applied to his small high-speed petrol engines by Daimler, whose patent did not expire until 1898. L. Funk had in 1879 patented a hot-tube igniting device with a sliding timing valve, but the essence of Daimler's arrangement is that the tube and combustion chamber are in constant communication.

A method of ignition largely used in oil engines is to feed the fuel direct into a chamber or 'hot bulb' connected with the cylinder, this chamber being first heated up for about ten minutes by an external lamp until a temperature is attained sufficient to ignite the compressed mixture; thereafter, the heating effect of the successive explosions maintains the unjacketed hot bulb at a sufficiently high temperature to ensure regular firing of the mixture, and the action becomes entirely automatic.

The Hornsby-Akroyd oil engine is a typical case. Fig. 184 is a section through the 'hot bulb' A and cylinder, and Fig. 185 shows the inlet and exhaust valves, also in section, placed in front of the 'hot bulb' A and cylinder section. The vaporiser bulb A is heated up by a separate lamp, the oil is sprayed in at the oil inlet B, and the engine is rotated by hand. The piston then takes in a charge of air by the air inlet valve into the cylinder, the air passing by the port directly into the cylinder without passing through the vaporiser chamber. While the piston is moving forward taking in its charge of air, the oil which has been sprayed into the hot bulb is vaporising and diffusing itself throughout the chamber, mixing, however, only with the hot products of combustion left over from the preceding explosion. During the charging stroke, therefore, air alone enters the cylinder, and the vapour formed from the oil is almost entirely confined to the hot bulb. On the return stroke of the piston air is forced through the somewhat narrow [neck *a* into the hot bulb, and there mixes with the vapour contained in it. At first the mixture is too rich in oil vapour to be capable of ignition. As the compression proceeds, however, more and more air is forced into the hot bulb, and just as the compression is completed the mixture attains the correct explosive proportions. The sides of the chamber A are sufficiently hot to cause explosion, and the piston moves forward under the pressure of the explosion so produced.

As the vaporiser bulb A is not water jacketed, and is connected to the metal of the back cover only by the small sectional area of cast iron forming the metal neck *a*, the heat given to the surface by each explosion is sufficient to raise its temperature to about 700–800° C. and keep it there.

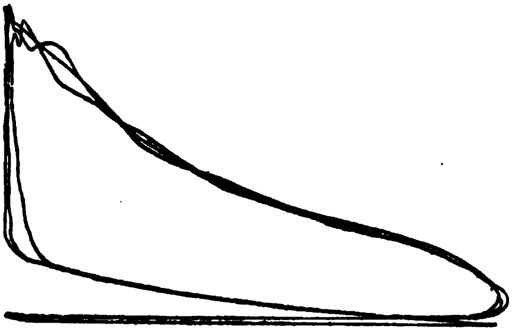


FIG. 186.—Cylinder Ignitions, Otto Engine

It is a peculiar fact that oil vapour mixed with air will explode by contact with a metal surface at a comparatively low temperature, and this accounts for the explosion of the compressed mixture in the hot chamber A, which is never really raised to a red heat. It has long been known to engineers conversant with gas engines that in certain conditions of the internal surface a gas engine will run and ignite with great regularity without incandescent tube or any other special form of igniter, if only some portion of the interior surface of the cylinder or combustion chamber be so arranged that its temperature is raised moderately; then, although that temperature may be too low to ignite the mixture at atmospheric pressure, yet when compression is complete the mixture will often ignite in a quite regular manner. Fig. 186 shows a series of diagrams taken from an ordinary Otto engine igniting in this manner without any special igniter, and it will be observed that the diagrams are very fairly regular. The author has noticed this peculiar fact in connection with one of his old engines. He placed a bolt, A (fig. 187), in the end of the piston B; this bolt was sufficiently long to project its head well into the explosive mixture; on starting the engine with the ordinary flame-igniting valve and running it for about fifteen minutes in the usual way, it was found that the flame-igniting arrangement could be then put out of action, when the engine would continue to run regularly, the mixture being ignited by the incandescent head of the bolt A, which projected into the explosive mixture and caused ignition just as the mixture became fully compressed. In this arrangement, however, the bolt A was found to attain a high red heat.

It is curious and interesting to note that with heavy oils ignition is more easily accomplished at a low temperature than with light oils. The explanation seems to be that in the case of light oils the hydrocarbon vapours formed are tolerably stable from a chemical point of view, but the heavy oils are very easily decomposed by heat with separation of the carbon and hydrogen; at the moment of liberation, the hydrogen, being in what chemists describe as the 'nascent' state, very readily enters into combination with the oxygen in the mixture. In this manner combustion is more easily started with a heavy oil than with a light one.

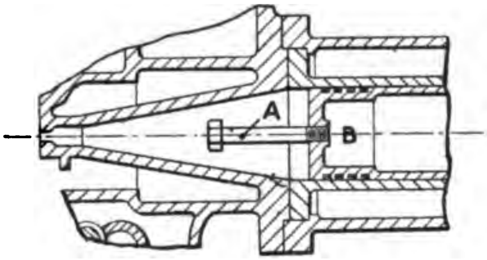


FIG. 187

The hot-bolt ignition just described has also been successfully adopted in some oil engines; Messrs. Clayton & Shuttleworth have patented an 'Automatic Igniter,' shown in fig. 188, of this type.

Referring to the illustration, which is a sectional plan through cylinder and vaporiser, C is the cylinder; P is the piston at the back end of its stroke; V is the vaporiser, fitted with its vapour valve V V, carried by a separate casing to admit of easy removal for examination when required; I is the igniter, formed of a small cylinder of cast iron containing a central steel needle surrounded by asbestos, which is kept in place by a perforated cap screwed on, and *h h* are ventilating holes; H is the holder which carries the igniter and permits of its easy withdrawal for renewal or examination.

Underneath the vaporiser is a starting tube, shown by dotted lines. This tube, by means of a 'Vesuvius' lamp, is heated to a good red heat before starting the engine, and serves to fire the first few charges until such time as the internal igniter becomes hot, when the lamp is at once withdrawn; all succeeding charges are then fired automatically from the igniter.

Messrs. Clayton & Shuttleworth state that in many cases the central steel needle can be dispensed with altogether, the asbestos alone causing the engine to fire quite regularly.

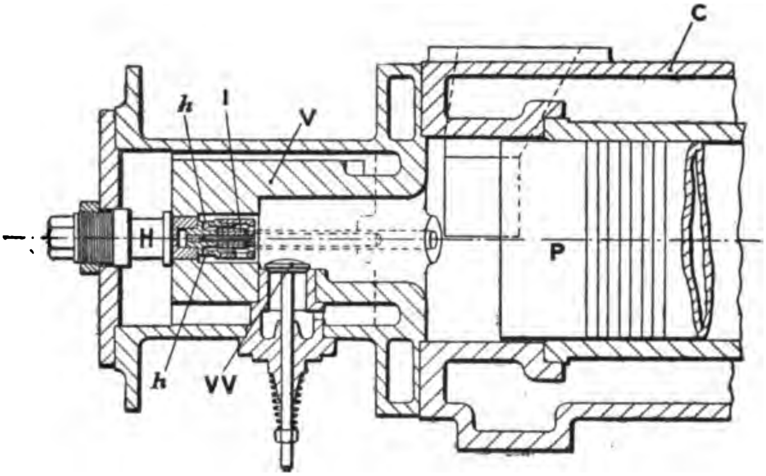


FIG. 188

The ignition of the two-cycle vertical engine of the American Oil Engine Co. is also automatically effected by a steel disc, or cheese-headed bolt, projecting into the combustion chamber.

Messrs. Tangye's crude oil engines are ignited by a hot bulb attached to the vaporiser, and maintained at the necessary high temperature to ignite the compressed mixture by the heat imparted by the explosions when once the engine is started.

In the Mietz and Weiss two-cycle oil engine, fig. 189, the oil is not injected into the hot bulb A, but is sprayed directly into the cylinder through a pipe B, and impinges against the heated lip C, thus becoming vaporised. It is claimed that in this way the vaporised oil is not mixed with the products of combustion remaining in A, and that a better mixture accordingly results.

An incandescent wire ignition device used by the author many years ago in experimental work may be conveniently mentioned here.

The current from a battery was utilised directly to heat a thin platinum wire. The difficulty of insulating is very slight. The tension being low, it is a matter of indifference whether the insulating material is wetted or not. The wire being constantly at a red heat cannot remain at all times in the cylinder, but is put into communication with it at proper times by means of a slide valve. Fig. 190 is a drawing of an igniting slide of this kind, as used by the author in experimental work. It acts very well. The screw 1 carries the rod 2, insulated by means of asbestos cardboard packed into the space and screwed down firmly by the gland 3. The other wire is screwed into the metal and so is in electrical connection with the body of the engine. One wire from the battery connects to any portion of the engine; the other is insulated. The platinum wire 4 is thus kept continually at a red heat, and the slide 5 suitably actuated causes the gases to be ignited to flow into the chamber containing the platinum spiral, through the hole 6, and so initiates the explosion.

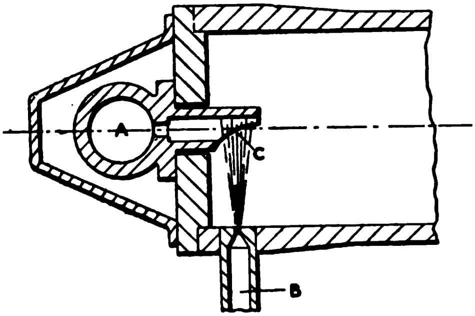


FIG. 189

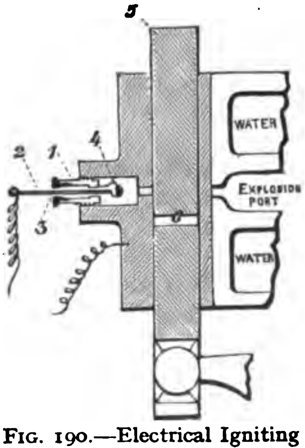


FIG. 190.—Electrical Igniting Valve (Clerk) Incandescent platinum wire

One precaution is necessary: the battery must not be too powerful. If the wire be heated by it to near its fusing-point, then the further heat supplied by the successive explosions may cause its destruction. It should show a good red heat and no more when open to the air; when closed up and in contact with the hot gases it will then become almost white hot; anything above this may fuse it.

The battery requires frequent renewal, and the method in this form is best suited for experimental work only.

Reference may here be made to the Diesel engine, in which no igniting arrangements whatever are necessary. The heat produced by the approximately adiabatic compression of the air to a pressure of about 500 lbs. per sq. in. suffices to cause instant ignition as the fuel oil is sprayed into the combustion chamber (see Chap. X.).

A recent design of engine for using crude and heavy oils by Messrs. Blackstone & Co., of Stamford, avoids the very high compression used in the Diesel engine. The piston compresses air alone into the combustion chamber and ' hot-pot ' or ' igniter bulb '—as it is termed by Messrs. Blackstone—to a pressure of about 150 lbs. per sq. in. A small spray of oil of constant amount is injected into the hot-pot at the moment of ignition, and immediately ignites, causing a flash of flame to be projected into the cylinder, where it meets a second spray of oil delivered directly into the combustion chamber, forming with the compressed air the working mixture ; this is, in its turn, immediately ignited. The power developed by the engine is regulated by varying the amount of oil sprayed into the combustion chamber at each working stroke.

The igniter bulb is brought to a red heat by a blow lamp at starting ; thereafter its heat is maintained by the explosions, and the ignition becomes then quite automatic.

3. CATALYTIC, OR CHEMICAL METHODS OF IGNITION

These are few, and none have become established in practice.

It is well known that spongy platinum possesses the property of causing spontaneous ignition in a stream of hydrogen or coal gas and air, when directed on it. Barnett in 1838 proposed to utilise this property to produce ignition ; in his arrangement the platinum is contained in a small cup affixed to the cylinder cover, and the compression of the mixture causes ignition by contact with it. Platinum does not, however, long retain this property, and moreover the action is too slow for use in any modern engine.

The proposal was, however, revived even so recently as 1896, and Messrs. Fahnenfeld & Wolfersgrun in Austria, and G. Lyon, of Cambridge, took out patents for improved igniters involving this property about that date. These have never come into use ; they are illustrated and fully described in the *Automotor Journal* for September 1900.

Finally the property of igniting spontaneously in contact with air possessed by phosphoretted hydrogen has been proposed as a means of ignition, a regulated quantity of the gas being admitted into the mixture in the combustion chamber at the moment desired for ignition; this suggestion has never been tried.

4. Electrical Methods

Improved electrical methods of ignition have practically displaced the arrangements above described except the incandescent tube, which is still used in small stationary gas and oil engines, and compression ignition as in Diesel engines of large dimensions. The electric spark has long been used by chemists to explode the contents of the eudiometer in which gas analysis is effected, and the idea of utilising its firing properties in the gas engine presented itself to inventors at an early epoch.

Philip Lebon, in 1799, proposed in his patent to use a machine operated by the engine for the production of an electric spark to ignite the charge. The use of the magneto machine as a means of producing a spark was also suggested by Stephard in 1850. Lenoir, about 1860, used electric spark ignition in his engine; a Bunsen battery, Ruhmkorff induction coil, distributor, and sparking-plug were employed substantially as so largely and successfully used over forty years later in the engines of automobiles.

Fig. 191 is drawn to show clearly the general arrangement. The Bunsen battery A generates the current, which passes by the wires to the coil B, from which the secondary current passes to the insulated electrodes of the plugs D, D by way of the high-tension distributor C. The negative high-tension pole of the coil is permanently connected to any part of the metal work of the engine; the sparking-plugs D, D are porcelain insulated as seen on a larger scale at E. The porcelain plug is firmly cemented into the brass nut 1, and the wire 2 which passes through a hole in the plug terminates outside in the connecting screw 3, and inside is bent over the end of the plug; the other wire 4 passes through another hole in the plug and is bent over in the inside, lying near the wire 2 but not touching it; this wire is carried through the side of the plug and is connected with the metal body 1. When the nut is screwed into position the one wire is in metallic connection with the cylinder of the engine, and the other is insulated from it.

The distributor C consists of an insulated metallic arm 1 rotating on the end of the crankshaft over the insulated ring 2, which is connected to the positive high-tension pole of the coil. Two insulated segments 3, 4 are connected by wires to the sparking plugs D, D; in rotating, the arm 1 comes alternately over 3, 4, and it is within sparking distance of the ring 2 as well as the segments; the sparks pass alternately to the segments and thence alternately to the opposite ends of the cylinder. The ebonite disc carrying the segments and ring is so adjusted that the spark begins to pass at either end just as the admission valve closes. If it passed too soon the explosion would occur before the admission valve closed. If it is passed too late, power is lost, because the piston is at its most rapid rate of movement and is reducing the pressure of the cylinder contents uselessly.

When all was in good order this arrangement worked regularly, but failure of the sparking-plugs at that time proved a frequent cause of trouble.

The design and constructive details in general of the ignition apparatus were, in fact, then so inadequate that this electric ignition did much to restrict the use of the Lenoir engine ; unless tended with great care and intelligence, constant troubles occurred.

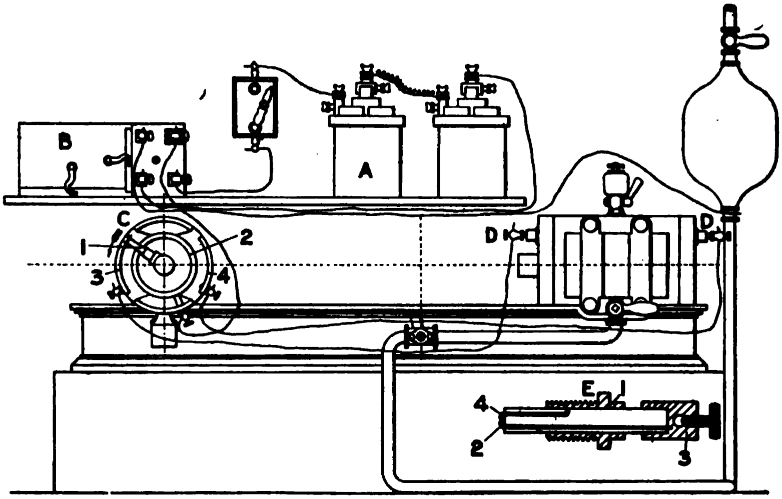


FIG. 191

In the early Priestman oil engines the ignition, following Lenoir's practice of thirty years earlier, was by Ruhmkorff induction coil and bichromate battery ; it gave a good deal of trouble.

The battery-coil method went entirely out of use for many years in internal combustion engines, but with improved manufacture and design was largely used in the small engines of motor cars between 1898 and 1908. Quite recently, also, the traction engines of Lehmbeck have empoyed the current from a Bosch dynamo to operate an induction coil with jump-spark ignition.

An interesting and important development of the coil and battery method of ignition is that due to Sir Oliver Lodge, F.R.S., and now well known as the 'Lodge system'; it is successfully used in all sizes, from the small motor of the automobile to the largest stationary gas engines; the quality of the igniting spark renders it of especial value where poor gas, e.g. blast furnace gas, is used. The Lodge method involves a 4-volt or 8-volt accumulator, induction coil, and sparking-plug; but the essential feature of the invention consists in the employment of a specially constructed induction coil; the following description by Sir O. Lodge renders the device and its action clear:

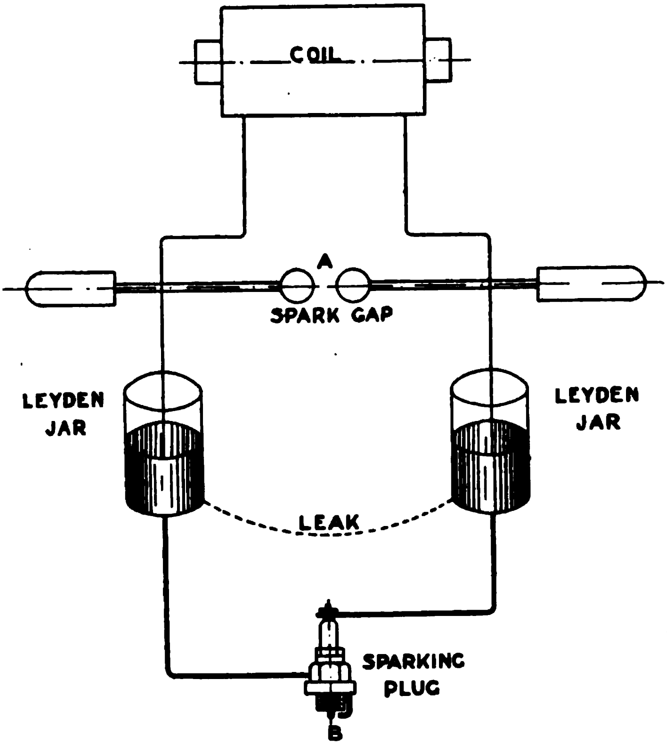


FIG. 192

'Fig. 192 shows the general principle of the Lodge igniter. The secondary terminals of the coil are not connected to the sparking-plug directly as usual, but are led to it through the intervention of a pair of coated insulators, or Leyden jars, with their outer coatings short-circuited by a leak or imperfect conductor, the object of which is to keep them always at the same potential, except at the instant of a sudden electric rush. Accordingly there is no strain thrown upon the leads or the sparking-plugs, whose terminals remain at the same potential up to the last moment when the two jars are full and overflow at A. At this instant everything is liberated, and with a rush of inconceivable rapidity, the jars empty themselves across A and round the complete circuit through the sparking-plug. No leak or imperfect conductor has time to exert any influence on the rush, which is over in the millionth of a second, but not before it has ignited the combustible mixture exposed to the sparking-plug.

'The rush is so violent that not only is dirt in the path blown away, but the electric momentum overshoots the mark, and the jars become charged up in the reverse way by the impetus; they then discharge again, and again are charged in the ordinary way, and so on many times, without the coil taking any further part in the action; its function is over when it has filled the jars to overflowing. The jar spark is a noisy white-hot spark of extreme suddenness, all over in the millionth of a second, which can be timed to occur with great accuracy.

'By this arrangement a discharge is brought about quite suddenly between points which except during the rush are completely inert, so that they might be handled with impunity, or placed under water, or clogged with dirt. But during the violence of the rush the dirt in the path is flung away, the water is burst through; the circuit is bound to be completed when the full discharge occurs at A. The A-spark is, therefore, a pioneer spark, which precipitates the sudden rush and causes the B-spark. The A-spark is in the coil-box under glass, open to inspection, and where it can be kept quite clean. The charge is prepared or generated by the coil, as usual, all the strain being thrown upon the clean gap at A, and directly this gap gives way the whole accumulated contents of the jars are suddenly emptied or exploded through the combustible mixture, the time of firing being accurately adjustable by the distributor which regulates the charging action of the coil.'

The imperfect conductor or leak between the jars was originally provided by a piece of damp wadding hermetically sealed in a small glass tube; in later car coil practice this has been omitted, the low conductivity of the wooden case of the coil being found sufficient. Also, in coils for car engines one 'jar' only is included; this takes the form of a condenser consisting of a flat plate of special glass, tin-foil coated on each side; one side is earthed, a lead from the other being led *viâ* the HT distributor to the central insulated electrode of the sparking-plug; see fig. 193.

For the ignition of large gas engines Leyden jars actually in 'jar' form are employed in conjunction with an 8-volt battery; the current consumption in these cases is about 0·5 ampères per cylinder, and is roughly proportioned to the number of cylinders.

Fig. 193 is a diagrammatic section of the Lodge car ignition coil, with the essential parts clearly indicated; the trembler and A-spark gap are seen at the top; the four terminals are completely enclosed by the bottom cover. The secondary winding is in sections, as in large induction coils, and not in layers as is so frequently the case in ordinary coils; in this way the electrical strain is more evenly distributed. For the 'secondary condenser' or 'Leyden jar' a special glass is used to withstand the electric strain. To economise the low-tension (battery) current as much as possible, the secondary winding of the

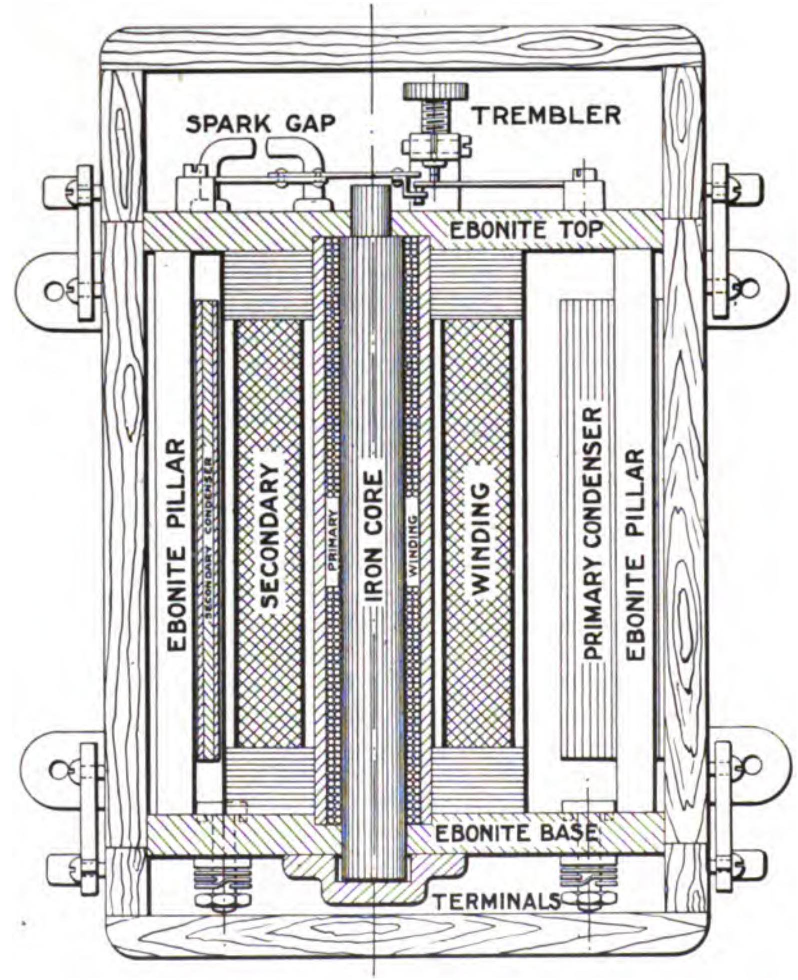


FIG. 193

Lodge coil is more than four times as long as that usually found in ordinary ignition coils; on a four-cylinder car engine the average current consumption is about 0·8 ampère. The trembler is easily and accurately adjustable, high-speed, and fitted with contact points of platinum-iridium of substantial size. The coil box is glass-panelled on top, so that the A-spark is visible during the running of the engine; the A-spark gap is made adjustable, and it is claimed that fouled sparking-plugs can be cleaned during the running of the engine by merely increasing the A-spark gap; the intensity of the B (igniting) spark is thereby so augmented that the obstructing matter is torn off the points of the plugs by the violence of the discharge.

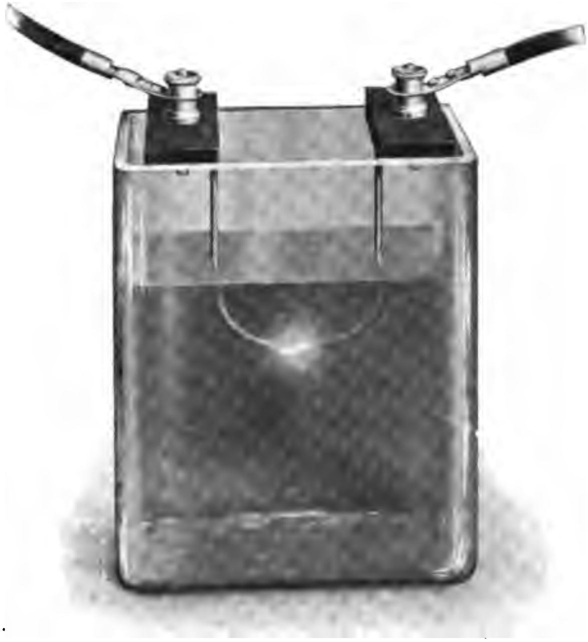


FIG. 194

In trembler coils coning and pitting of the platinum contact points usually soon occurs, the metal being transferred from the positive contact and built up on the negative. Messrs. Lodge Bros. eliminate this source of trouble almost wholly by the use of a reversing switch whereby the direction of the battery current through the coil is changed each time the switch is used.

The B-spark is an extremely rapidly oscillating Leyden jar discharge, the electrical surgings causing, momentarily, enormous voltages to be attained. So sudden and intense is the rush—the whole duration of the discharge being only about the one-millionth part of a second—that ordinary obstacles are broken down. As the author has himself witnessed, the discharge will even occur under water; fig. 194 shows a Lodge B-spark occurring under water; Messrs. Lodge state that in this particular case a powerful coil was used and several glass vessels were broken by the shock of the spark before a photograph could be obtained. The metal plates on the top carrying the wires were also forced up, and it was found necessary to clamp them very firmly to the glass.

A valuable feature of the Lodge method is that by its use effective double ignition can be obtained in the engine cylinder; two sparking-plugs connected in series and placed diametrically opposite one another are used; the rapidity of ignition may thus be increased, tending to improved engine efficiency. The author has used the Lodge ignition on the experimental engine in his laboratory with excellent results; a single plug only was employed.

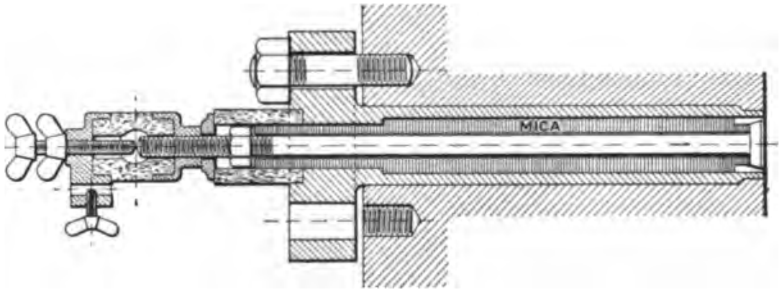


FIG. 195

The Oechelhauser engines constructed in this country by Messrs. Beardmore at Glasgow employ the Lodge ignition. In a single-cylinder engine two plugs are used in series; with double-acting, tandem engines there are four plugs per cylinder, each igniter operating two plugs simultaneously. The low-tension (primary) current for the igniters is furnished by an 8-volt accumulator which, when practicable, is connected up to an electric main so as to remain constantly charged.

The sparking-plugs are of special design, and a section is shown in fig. 195. The plug consists essentially of a mica-insulated central spindle terminating inside the cylinder in a disc, and contained in an outer metallic case between the inner end of which and the disc a narrow annular space is left, across which the igniting B-spark occurs. The annular spark-gap, about 0·03″ wide, offers no isolated points that may become incandescent during the working of the engine, and thus cause pre-ignition. At the outer ends of the plug there is an adjustable spark-gap, clearly indicated in the figure, the function of which is to separate the capacity of the leads from that of the central spindle of the plug, and thus prevent a sparking discharge occurring between the central spindle and the cylinder casting.

An advantage of this system is that, when once adjusted, the firing does not become late after a time due to wear at the sparking point, as is the case with the low-tension magneto system to be next described.

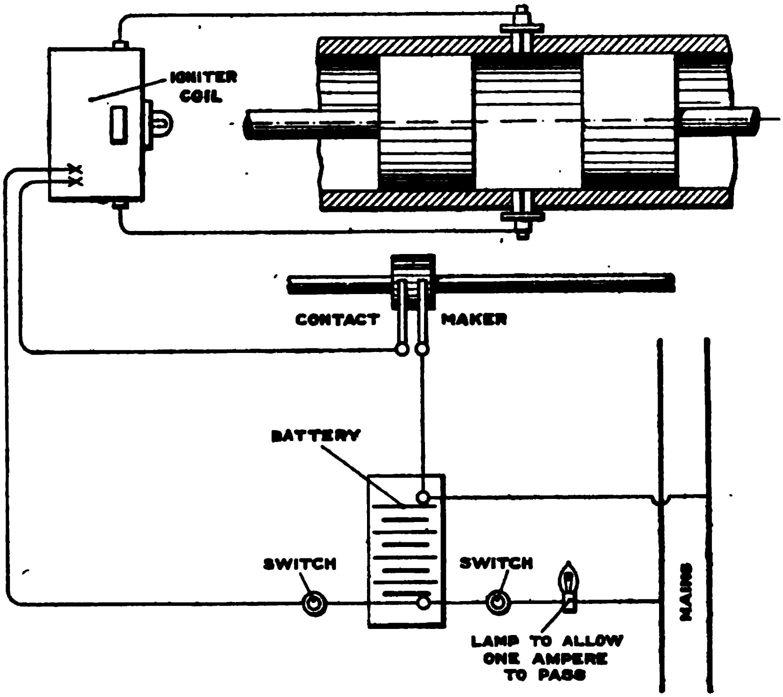


FIG. 196.

To ensure the occurrence of an ignitible mixture near the plug, an annular governor-controlled slide is provided in the Borsig-Oechelhauser engines, fitted outside the inlet ports, and so arranged as gradually to close the ports remote from the plug as the load on the engine is diminished. Finally, at light load, a few ports only—and these in the vicinity of the plug—remain open for the admission of the mixture ; regular and sharp ignition is thus obtained.

In the accompanying diagram, fig. 196, the wiring arrangement is exhibited; this is simple and self-explanatory.

The Lodge system is used also in the engines of Ehrhardt & Sehmer; Richardson, Westgarth & Co.; the Campbell Co.; Messrs. Hindley; the Fordingham Steel Co., &c. In the concluding section of this chapter, on 'Sparking-plugs,' will be found illustrations of two recent designs of Lodge plug for large gas-engine ignition.

There is a very pronounced tendency on the part of users of internal combustion engines, especially for automobile work, to dispense with accumulators whenever possible, and it is probably for this reason that igniting systems involving their use have not of recent years kept pace with the magneto methods now so very extensively adopted.

In the early stationary two-stroke Benz engines jump-spark ignition was employed, the primary current for the induction coil being supplied by a dynamo rope-driven from the engine. The dynamo and coil were kept constantly running, and a small external rocking arm on the sparking-plug (fig. 197) shorted the high-tension circuit except at the instants required for ignition of the mixture.

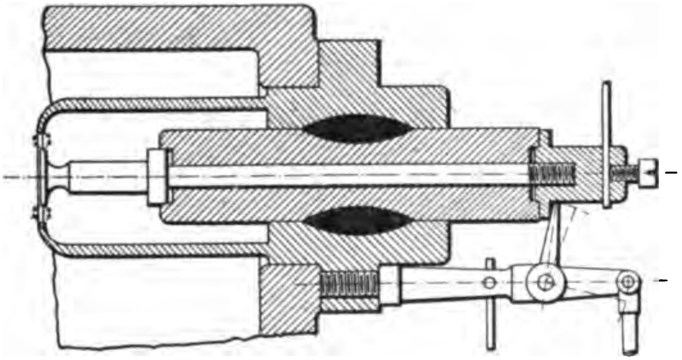


FIG. 197

The early de Dion motor tricycles were fitted with a similar mode of ignition. A step in advance was next made by dispensing altogether with the induction coil, and effecting the ignition by means of a specially designed electrical machine with a rocking armature and break contact within the combustion chamber of the engine. This is known as the low-tension magneto system, and is a method at present largely used in stationary gas engines. In the earlier designs the armature itself was rocked, but its inertia was found to cause wear and trouble in the bearings and gear. Messrs. R. Bosch, of Stuttgart introduced a type, now very widely used, in which the Siemens I-section armature remains stationary in the magnetic field between the poles of the powerful permanent horse-shoe field magnets, while a split soft-iron sleeve, partly enclosing the armature, is alone caused to rock, thus deflecting the lines of force in the field, and causing a current to circulate in the armature windings and around the firing circuit.

Usually a cam or eccentric on the half-speed shaft is so connected up with the magneto that the rocking sleeve is slowly rotated through a certain angle and then suddenly released by a trip device, when it is instantly swung back to its initial position by a strong spring provided for the purpose.

This arrangement avoids the earlier inertia troubles, gives a good firing spark, and enables the current to be taken from a stationary armature, thus obviating the necessity of collecting it by brush contacts from moving surfaces.

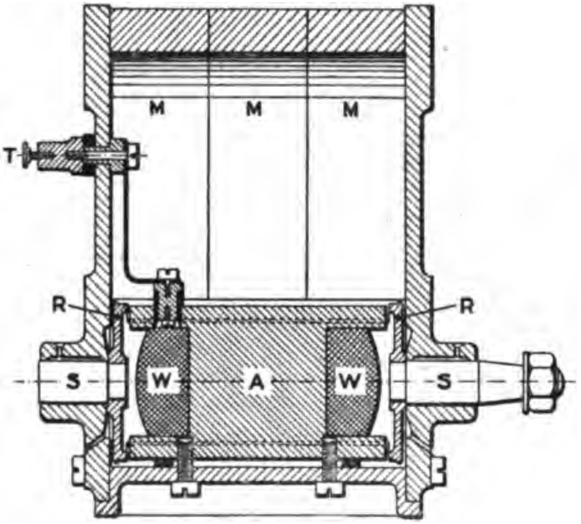


FIG. 198

One end of the firing current is connected to the frame of the engine, and the other to the insulated spindle of the make-and-break plug on the cylinder.

Fig. 198 shows the essential parts of the machine in diagrammatic form ; M M M are the permanent field magnets ; A is the fixed armature core of I-section ; W the armature winding ; R R are the ends of the rocking shield carried on the spindle S S ; T is the insulated terminal of the armature winding, the other end being ‘ earthed ’ to the engine frame.

A type of make-and-break plug very generally used is shown in section in the accompanying diagram, fig. 199. The device is carried in a casting A A (sometimes water-cooled) attached to the combustion chamber of the engine by two studs and nuts, thus facilitating easy removal for adjustment or repair. This is an important point practically as it is very common for difficulty to be experienced in starting the engine from 'all cold' owing to the deposition of moisture which always then occurs in the cylinder, causing failure of the ignition at contact-breaking. The remedy usually resorted to is to remove the whole igniter casting and warm it thoroughly; starting is then effected without further difficulty.

In fig. 199 C C is the rocking spindle, often made with a conical seating at D to ensure gas-tightness—carrying at its inner end a short lever whereby the make-and-break of contact is effected with the insulated spindle B B. C C is actuated by gear from the engine through the crank E.

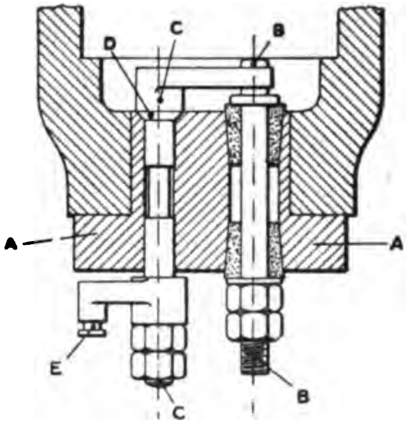


FIG. 199

The sparking current, produced at the instant of breaking contact, is of high voltage, and the insulation of the fixed spindle B B has required careful designing. Asbestos, soapstone, mica, porcelain, and enamel have all been used. The material employed must be not only a good insulator, but also capable of withstanding a high temperature and the mechanical effect of the constant hammering action of the break-contact lever C on the end of the spindle.

To prevent rapid burning away of the make-and-break contacts various metals and alloys have been used. The insulated spindle of the plug is usually of steel sheathed at the 'anvil' end with some special metal or alloy; similarly the lever arm in the end of the rocking spindle C C is of special material. Nickel, nickel-steel, 'Durana' metal, and 'Casalloy' have been much employed. Platinum alloys have also been used, but are very expensive and usually somewhat troublesome to replace.

In fig. 200 the whole mechanism is diagrammatically shown. A trip lever on the half-speed shaft turns the shield relatively slowly through the angle indicated and releases it at the moment of ignition, when it flies back suddenly to its initial position under the action of the two stiff helical springs shown. Simultaneously contact is broken at the firing plug in the combustion chamber, and the igniting spark leaps across the gap.

When an engine is required to run at full speed it is necessary to advance the ignition. The instant of the quick-return motion of the rocking sleeve or armature of the magneto and that of the contact-break in the ignition plug should, therefore, in order to obtain the best results, be both adjustable relatively to the piston position. So long as the rocking sleeve, or armature, and contact breaker are operated by the same springs the instants of maximum current intensity and breakage of the contact coincide. When the rocking sleeve or armature is actuated from the half-speed shaft—as is usually the case in high speed engines—it is not practically an easy matter to so arrange that the instant of maximum current intensity is also that of the firing spark. Often, therefore, the contact-breaker in the plug is alone made adjustable, with the result that the spark does not occur quite at the moment of maximum current intensity when the ignition is advanced; practically, however, it is found that the firing is in general not much impaired, and accordingly it is only where very special reliability of ignition is necessary over a wide range of speed that arrangements are adopted wherein both the rocking shield and the plug make-and-break are adjustable. Automatic adjustment of the ignition by the engine governor has so far presented considerable difficulties which have not yet been fully overcome.

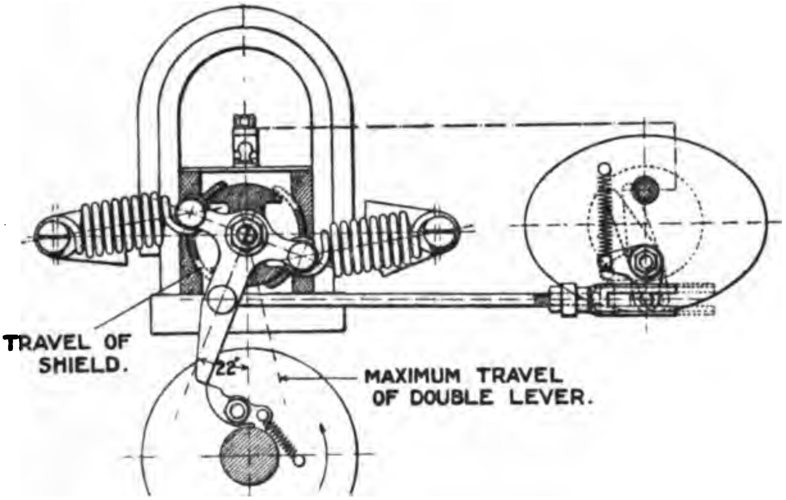


FIG. 200

The National Gas Engine Co., of Ashton-under-Lyne, employ low-tension magneto ignition in all their larger engines.

Each cylinder is served by two magneto machines and two separate ignition plugs placed diametrically opposite one another in the combustion chamber; the general arrangement is clearly indicated in fig. 57.

A crank pin, A, of 1 in. throw, projecting from the end of the half-speed shaft, actuates the magneto lever arm J, through the rod B carrying a trip, D, at its extremity. The rod B is borne by a swinging guide, C, supported on a small eccentric at F, whereby it may be fixed in a higher or lower position, and thus vary the time of release of the magneto lever arm J, and also the moment of breaking contact in the plug K—which is directly operated also from the rod B, as indicated in the figure.

A tension rod, E E, passes across to the other side of the cylinder, and simultaneously operates the second magneto and ignition plug.

The quick return of the rocking shield, or armature, of the magneto is caused by the stiff helical spring H.

The actuating gear is here simple and effective; it permits the time of ignition to be varied while the engine is running, and provides that the two plugs shall always fire simultaneously. As the plugs are operated directly from the magneto lever arms without intervening link-work, they may be at any time withdrawn for examination without disturbing the adjustment of any gear; this is an important practical advantage.

The lower figure shows an ignition plug in section; the constructive details are clearly indicated. M is the insulated spindle, bushed at its inner end or 'anvil' with a nickel or nickel-steel bush. N is the movable spindle with a 'hammer' of the same material fixed in its enlarged inner end. The gas-tight metal-to-metal joint is the narrow conical ring L L. The plug casting is attached to the cylinder by the two studs and nuts shown in the upper view. The casting is well ventilated, and the insulation of the spindle M carefully designed to resist the racking effect of the expansion and the hammering action of the contact-breaker. In engines using producer gas and blast furnace gas low-tension magneto ignition has proved specially serviceable and is now very extensively adopted. To guard against failure and obtain more rapid and uniform combustion, duplicate ignition gear has for some years been fitted by several makers to all engines of more than from 80 to 100 horse-power per cylinder.

In one arrangement by Messrs. Koerting—who have used duplicate ignition during the past ten years—the two low-tension magnetos are placed close together on the same bracket with their trip levers connected by a short coupling rod. By suitable link-work the movable spindles of the ignition plugs are actuated from one of these trip levers, the other receiving motion from trip-gear driven by the half-speed shaft of the engine. One ignition plug is placed in the cylinder end, and the other in the side of the cylinder.

In some of the Nürnberg engines the ignition is of the low-tension type, but instead of the magneto machine and attendant driving gear, a battery alone is employed.

The sparking-plug or 'igniting block' has affixed to it an electro-magnet which when the current is established causes a hammer piece to swing suddenly over. In its motion it trips the rocking spindle of the sparking-plug and thus causes the sudden break necessary to fire the cylinder mixture. A wipe-contact on the half-speed shaft distributes the current to the several plugs. The consumption of current is said to be economical, a tandem engine requiring only about 0·5 ampères at 60 volts.

Messrs. Bosch have also recently introduced a low-tension magneto ignition using a specially designed magnetic plug constructed on the 'Honold' system for single-cylinder stationary gas engines of all kinds running at not exceeding 250 revolutions per minute. The magneto is of the usual spring-controlled rocking-armature type, the armature oscillation being through an angle of 30°. One end of the armature winding is earthed to the armature core, while the other is carried to an insulated terminal and thence to the magnetic ignition plug. A section of this plug is shown in fig. 201. The interrupter lever 1 is carried on the steel knife edge 2, and rests also in the contact block 7; a spring 3 presses on the back of the interrupter just inside the knife edge and causes its return after the current break. The coil body 4 is screwed at its lower end to the core 5 of the plug, thus forming a complete magnetic circuit, the armature being the upper part of the interrupter lever.

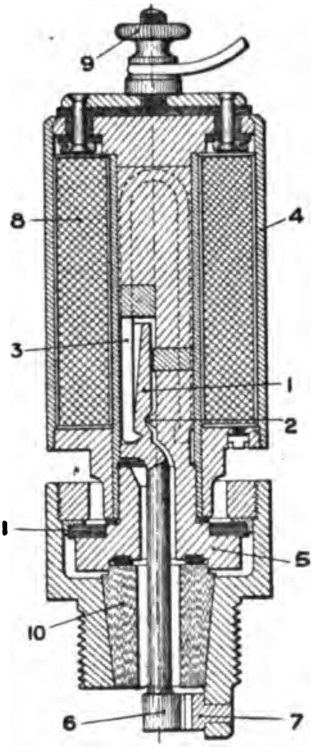


Fig. 201

The contact pieces 6 and 7 (see also fig. 202) are mounted upon the interrupter lever and plug body respectively and project well into the combustion chamber; these are kept in contact by the spring 3. Contact is broken at the firing instant by the current from the magneto passing through the plug coil 8; the passage is from the insulated terminal 9 round the coil; thence to the coil core and interrupter, which are insulated from the plug body by a steatite cone, 10, and mica washers, 11. The current thus passes across the contact between 6 and 7, and returns by way of the engine frame to the magneto armature; the core becoming magnetised attracts the interrupter lever, causing it to rock on the knife edge 2 and thus break the contact between 6 and 7; contact is remade by the action of the spring 3; a large, white, flaming spark is produced at break.

Care must be taken to keep the interrupter free from gummy oil deposit, which may prevent its prompt action; this lever is readily accessible for cleaning with the construction adopted.

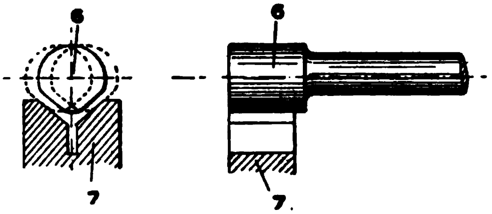


FIG. 202

The contact pieces are of special form; fig. 202 shows the disposition of these clearly; the cylindrical end 6 of the rocking interrupter lever seats itself in a V-groove in the 'anvil' 7; the interrupter has a small amount of side play which is found to lessen the liability to failure of the ignition by sooting up.

The 'Honold' plug must be fitted to a well-cooled portion of the cylinder casting, and wherever practicable should stand in a vertically upright position. The plugs require cleaning about once a week; the coil body is unscrewed and the rocker freed from gummy oil by paraffin or petrol; if soot is suspected on the contact breaker the whole plug must be removed from the cylinder. Messrs. Bosch also supply 'Honold' plugs and suitable *continuous* rotation magnetos for 3, 4, and 6-cylindered petrol engines of the automobile type.

Between the usual low-tension magneto and the high-tension magneto with rotary armature may be placed an igniting system described as the Bosch 'arc light' magneto system.

The apparatus consists of a magneto externally generally similar to the usual low-tension magneto, and having a rocking armature with springs operated by the usual trip gear. The armature, however, has a secondary winding in addition to the primary; one end of the secondary is 'earthed' to the primary, and the other is connected, through an insulated collecting ring and carbon brush, with the insulated central stalk of an ordinary fixed-part sparking-plug.

With this magneto the timing of the ignition is varied, as in the low-tension system, by providing some means of adjustment in the link-work actuating the trip lever.

This system obviates the use of igniting plugs having movable parts. The igniting spark is a hot, arc-like discharge capable of firing poorer mixtures than the usual jump-spark.

These machines are constructed for use with stationary internal combustion engines having a compression pressure of about 90 lbs. per sq. in. above atmosphere, and requiring not more than about 250 ignitions per minute.

The endeavour to avoid the troubles inherent to the use of ignition plugs with movable parts coupled with the remarkable degree of efficiency attained in the present high-tension magneto machines with continuously rotating armatures, fully described in the next section, render it very possible that the near future may witness the adoption of this latest mode of ignition, already practically universal in the engines of motor vehicles of all sizes, by makers of large stationary gas and oil engines also.

Messrs. Bosch have already produced a dual ignition system for large stationary gas engines, including a specially large and substantial design of H.T. rotary magneto and sparking-plug. The plug is illustrated and described in the special section at the end of this Chapter. The magneto, described as 'Type S.H. 4' for four-cylinder engines, is of the usual pattern but considerably larger; sparking occurs, in air, across the plug gap at so low an armature speed as 35 revolutions per minute; the ignition can be adjusted through an angular range of 30° on the distributor axis.

This H.T. magneto system is now being experimented with by several makers of large stationary engines. A diagram of connections for a four-cylinder engine on this system is given in the accompanying fig. 203.

The simplicity of the rotary drive, complete absence of oscillating link-work, facility of varying the time of ignition, and absence of moving parts in the igniting plugs are the very valuable features of this system.

In the above account reference has been made to the most noteworthy of the many ignition arrangements which have, from time to time, been proposed and used in stationary internal combustion engines. Ignition in the small fast-running engines used in automobiles has presented a special problem to which the next section of this chapter is devoted.

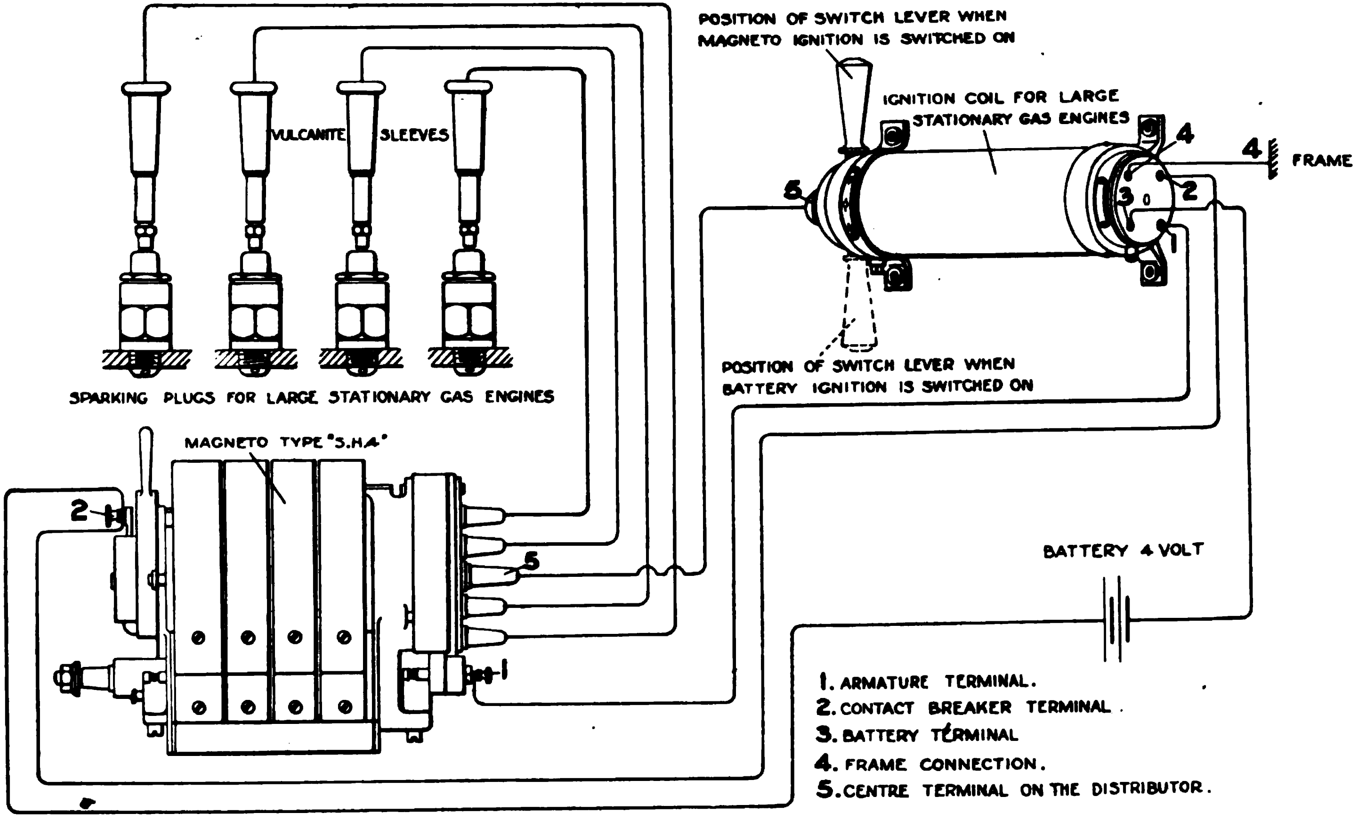

POSITION OF SWITCH LEVER WHEN MAGNETO IGNITION IS SWITCHED ON
IGNITION COIL FOR LARGE STATIONARY GAS ENGINES
4
4
FRAME
5
3
0
2
1
POSITION OF SWITCH LEVER WHEN BATTERY IGNITION IS SWITCHED ON
VULCANITE
SLEEVES
SPARKING PLUGS FOR LARGE STATIONARY GAS ENGINES
MAGNETO TYPE "S.H.4"
2
5
1
BATTERY 4 VOLT
1. ARMATURE TERMINAL.
2. CONTACT BREAKER TERMINAL.
3. BATTERY TERMINAL
4. FRAME CONNECTION.
5. CENTRE TERMINAL ON THE DISTRIBUTOR.

FIG. 203

SECTION II

IGNITION IN PETROL MOTORS

It will be readily realised that ignition in the small engines of motor cars and boats running at speeds ranging from 600 to 1500, or even more, revolutions per minute under constantly changing loads and in very many cases driven by non-technical persons, has proved a matter of special difficulty requiring much attention on the part of the designer.

In the early car engines, up to about 1900, tube ignition, usually without a timing valve, was almost universal. The tubes were of nickel or platinum maintained at a red heat by means of external lamps using paraffin or petrol.

Fig. 204 shows the usual arrangement, the illustration being of an early Daimler motor; the platinum tube and petrol-burning heating lamp I are clearly visible; the tube is fitted close to the automatic inlet valve F to ensure a rich and readily ignitible mixture entering it during the compression stroke of the engine.

Means were sometimes provided whereby the heating lamp could be moved in both a vertical and horizontal direction, so that the requisite degree of heat could be maintained in the tube and also that the heated zone could be caused to occur nearer the cylinder end of the tube when faster running of the engine was desired.

This mode of ignition was adopted even in the very small engines of motor cycles; in a Report of the Paris-Nantes motor race, September 1896, it is stated that owing to the high wind prevailing during the run the tube-heating lamps of the cycles were frequently extinguished. In his address to the Liverpool branch of the Self-Propelled Traffic Association in 1896, Sir D. Salomons referred to the constant trouble arising from the heating lamps being blown out in windy weather. Notwithstanding this, and the further difficulties from cracked or burst tubes, the system continued in use in default of anything better until about the end of 1899, when the practical advantages of the coil and battery method began to attract general attention.

The above-mentioned troubles and the risk of fire attending the use of naked lights on a petrol engine, led to a speedy abandonment of tube ignition, and many engines of cars thus fitted were converted to the electrical method. This method was in almost universal use from about 1899 to 1906; in its essential features it did not differ from the ignition used by Lenoir in 1860, as already described, the improvements being mainly in points of constructive detail.

Though now completely superseded, in its turn, by the high-

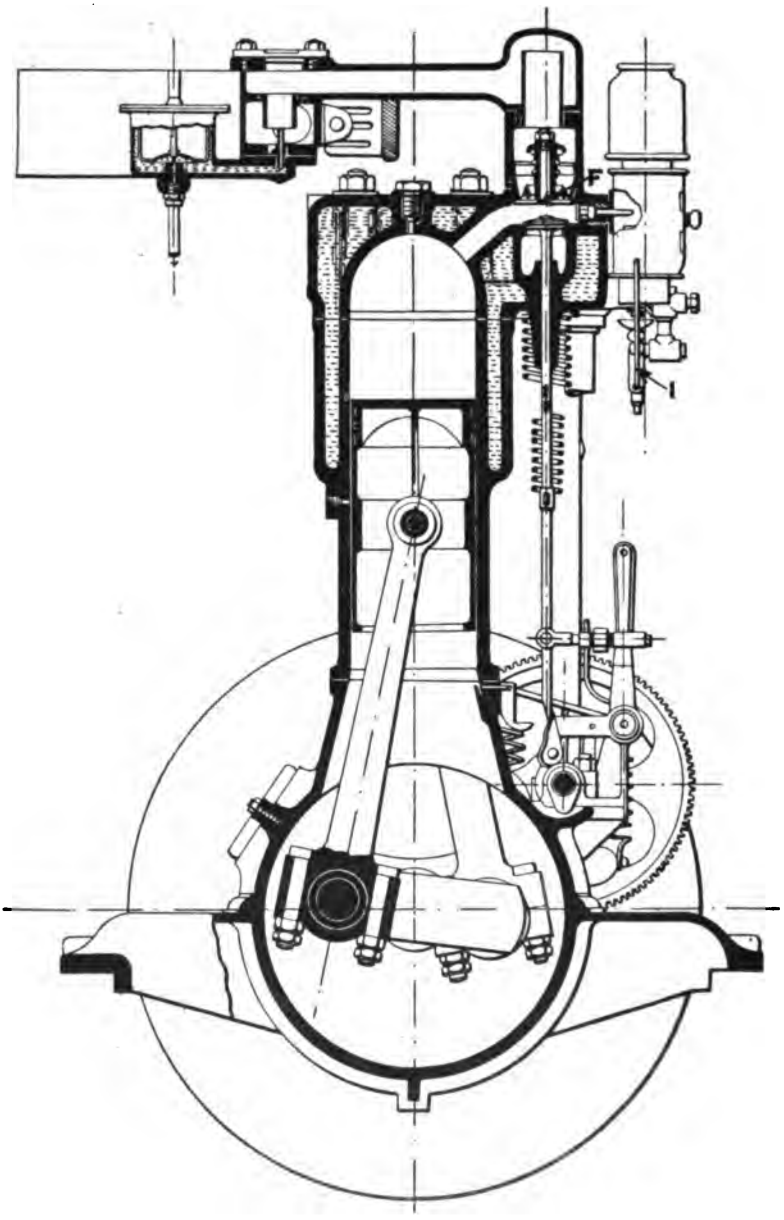


FIG. 204

tension magneto system, the coil and battery rendered valuable service for several years, and when kept in proper adjustment proved quite satisfactory in ordinary cases.

It possessed the important advantages of facilitating engine starting, and of enabling the instant of ignition to be readily varied at will.

As Dr. Watson has very clearly pointed out, however, (Cantor Lectures, 1910) the trembler possesses certain inherent defects which militate against fast and even running of an engine using this form of coil. Thus he found that a good type of trembler when well adjusted would average about 200 breaks per second, i.e. gave one break in 0·005 second. Now with an engine running at 1200 revolutions per minute, and having a contact sector B of 20° on the distributor cam (fig. 208), the battery current is only completed during $\frac{1}{180}$th (= 0·0056) of a second; hence it is clear that for this and higher speeds of rotation there is only time for the trembler to break the battery circuit once, giving a single firing spark only, just as if a non-trembling coil were used. With a weak current the trembler may fail to act owing to the magnetisation of the core being insufficient to attract the vibrating armature enough to cause a circuit break, and this will occur also at very high speeds owing to the duration of the completed battery circuit by the contact sector being so short that the battery current has not time to rise to more than a fraction of its steady value before break occurs. Professor Springer's oscillograph records, cited by Dr. Watson, indicate clearly that even after an interval of 0·0065 second from the epoch of 'make' the primary current was still perceptibly increasing.

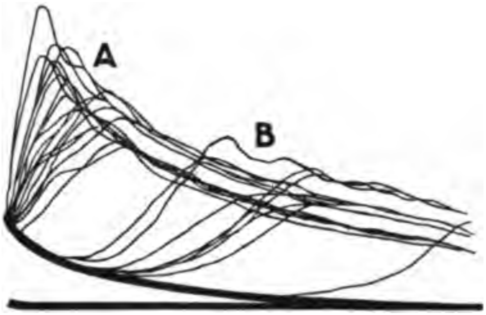


FIG. 205

Accordingly in such cases the trembler becomes inoperative and the make-and-break is then effected by the distributor contact sector B (fig. 208); this causes the ignition suddenly to become late, since the firing spark occurs at *break*, and is thus later when the sector governs the ignition than when the trembler is in action.

The result is very clearly indicated in the annexed fig. 205; as Dr. Watson points out, the diagrams are divisible into two groups, A and B, group A corresponding to the trembler acting, while in group B the trembler is inoperative, and the ignition retarded through being then timed by the distributor sector alone.

Non-trembling coils were also largely employed by some makers, notably Messrs. de Dion, Bouton & Co., whose engines have always had a high reputation for speed and power. As the effectiveness of the secondary (firing) spark depends upon the suddenness of the interruption of the primary current, special make-and-break devices were employed in conjunction with non-trembling coils. Fig. 206 shows the well-known early form of the de Dion contact-breaker. A is a cam on the half-speed shaft notched as shown; the hardened steel end B of the spring contact blade C drops into this notch once in every revolution of the cam, thus completing the battery circuit through the platinum points D. The device was mounted on a plate H H, commonly of red vulcanised fibre in small engines, borne on a bearing co-axial with the half-speed shaft K. The time of ignition was varied by giving H H an angular displacement around K by means of the hand control rod G.

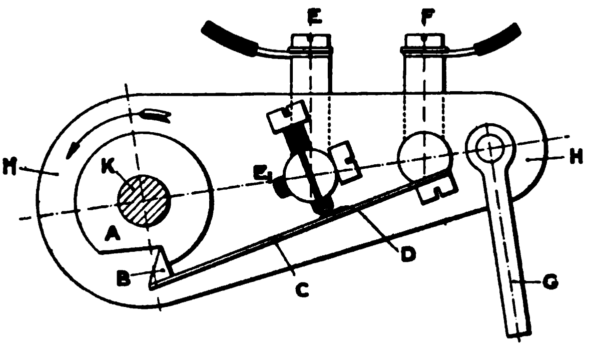


FIG. 206

The adjustment necessary to obtain good running with this arrangement usually involved considerable care and patience; breakage of the spring contact blade was a not infrequent trouble; and it was often difficult to start the engine by hand on account of the parting of the platinum points at 'break' occurring with comparative slowness. The timing of the ignition was also dependent upon the distance the steel point B dropped into the notch of the cam A.

An improved design, giving a much more sudden break to the primary current, is shown in fig. 207. As the cam A rotates, the spring B is lifted so that a platinum stud on its upper surface comes in contact with a stud on the lower surface of the spring C, thus completing the primary, or battery, circuit. When the cam rotates somewhat further the springs B and C both commence to descend, and acquire some velocity when the spring C strikes the pin D and the circuit is abruptly broken. In this way a quick break is secured without the necessity for making the sides of the cam very steep, so that the wear of cam and contact block is reduced to a minimum.

Owing, however, to the difficulty frequently experienced in starting the engine on a non-trembling coil, this was often replaced by a coil with trembler, although, when once started, faster running and more regular firing were obtained with the non-trembling coil for the reasons indicated above.

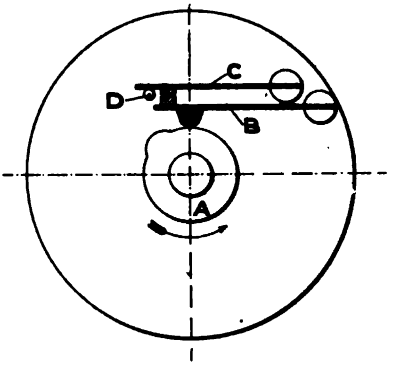


FIG. 207

Lenoir used primary batteries, but in later years the secondary battery or accumulator has been most generally employed on account of its smaller bulk and convenience in recharging. Some makers, including Messrs. de Dion, have, however, for long used 'dry' primary batteries in their ignition arrangements. These were of large capacity on account of their high internal resistance, and four were generally employed, connected in series. The dry battery is particularly suitable for use with a non-trembling coil, as Mr. Strickland has pointed out in his work on Petrol Motors.

With coil ignition a four-volt battery was most often employed; occasionally 6 volts were used. The mean current consumption with a trembler coil, when the action of the trembler is practically continuous, need not exceed 0·5 ampères at 4 volts with good adjustment; small current consumption is desirable, partly because the accumulators require recharging less frequently, partly because the platinum contacts do not burn away so rapidly; with a bad adjustment the current consumption may rise as high as 3 ampères.

The 'wipe' contact distributor, or 'commutator' was used in all ignition systems employing induction coils with tremblers for the 'making' of the primary current. In its simplest form for a single-cylinder engine it consists (fig. 208) of a fibre disc A mounted on the half-speed shaft C and carrying a brass contact sector B connected by a set screw with the driving shaft, as shown. D is a spring blade carrying the hardened steel contact-block E, which presses against the fibre disc A and is fixed at its other end to the insulated bracket H carried on the holder J. To the insulated blade D one lead of the battery-coil circuit is attached, the other being 'earthed' to the engine frame; the current is established when the brass sector B in the course of each revolution comes into contact with the block E. The ignition is advanced or retarded by means of the control rod F.

For a two- or four-cylindered engine the same fibre disc A suffices, but it is of course necessary to have two or four spring-blades D, arranged around the circumference of the disc. For the ordinary two-cylinder engine with cranks at 180° the spring-blades of the wipe contact are only 90° apart, but for a four-cylindered engine the disposition of blades is symmetrical.

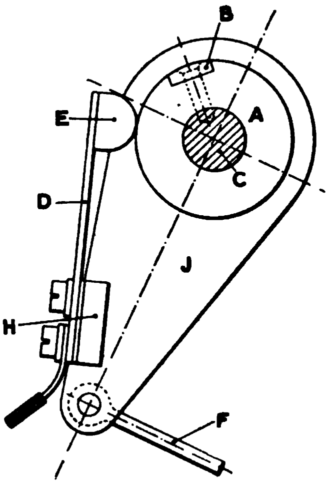


FIG. 208

This simple device works very well in practice; the fibre disc requires occasional trueing up, and the contact blocks E wear to a 'flat' by the rubbing of the grit-holding surface of the fibre disc A against them, and must be ground from time to time to diminish the length of surface of contact.

Messrs. Lodge Bros. employ an improved mode of construction involving two metal 'brushes' bearing on a practically continuous steel ring divided like the commutator of a dynamo. The insulating strips are made narrower than the brushes, so that these latter rub always on a smooth metal surface. The required length of contact is obtained by altering the angular distance apart of the brushes.

The diagram, fig. 209, indicates the essentials of the modern coil and battery system; for simplicity the case of a single-cylinder engine only is shown.

A is the battery or accumulator, usually of four volts, and averaging 40 ampère-hours capacity. As constructed for engine ignition, a 40 ampère-hour four-volt cell is about 4 in. by 4½ in. by 6½ in. in external dimensions; a second cell is generally fitted as a stand-by in addition to the working cell. The plates are contained in transparent celluloid casings, rendering the contents of the cell visible.

B is the induction coil, consisting of a soft-iron-wire core wound with a primary winding consisting of a few turns of thick wire, and a secondary of many turns of very fine wire. C is the interrupter or 'trembler,' operated by the electromagnetic action of the coil. D is the 'wipe-contact' distributor whereby the battery circuit is established at the required moment of ignition. E is the condenser to minimise sparking at the trembler—and consequent burning of the contact points—and also improve the action of the coil. F is the sparking-plug in the engine cylinder. One end of the secondary circuit is connected to the insulated central stalk of this plug, and the other to the primary circuit and 'earthed' with it to the engine frame, as shown diagrammatically at H. G is a switch in the battery circuit.

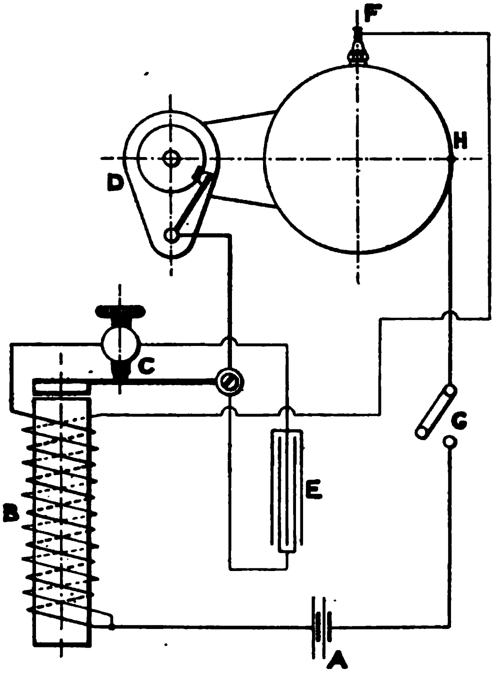


FIG. 209

To render less frequent the necessity of recharging the accumulators some engines were fitted with a small continuous-current dynamo, driven by belt or friction from the engine, and so arranged as to be continuously charging the accumulators while running; a mechanical cut-out, usually of the centrifugal governor type, prevented reversal of the current through the armature on slowing down or stopping the engine.

In other cases again, either the dynamo or the accumulator could at will be connected with the induction coil through a two-way switch. To start the engine the accumulator current was employed; so soon as speed was attained the accumulators were cut off and the dynamo switched on to the coil. Thus the accumulators were but little used, and accordingly rarely required recharging.

This arrangement soon proved unsatisfactory in practice. The dynamo voltage increased with the engine speed, and break-downs of the induction coil became not infrequent in consequence; on the other hand, on slowing down the engine the voltage would become insufficient to produce ignition, and an involuntary stoppage would occur unless the accumulators were promptly switched into action.

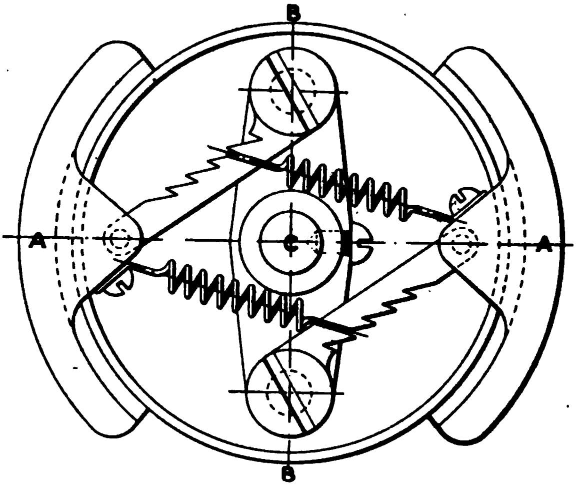


FIG. 210

Attempts were made to limit the rise in voltage, a favourite device being to drive the dynamo by belt from the engine through an adjustable friction pulley on the armature spindle set so as to slip when the armature attained a pre-determined maximum speed, and thus prevent break-down of the coil. Fig. 210 shows the belt pulley on an 'Apple' dynamo thus fitted; the fibre-shod spring-controlled centrifugal pieces A A are attached to the armature spindle C, the disc B B and attached belt pulley behind it being loose on this spindle.

As the speed of rotation of the dynamo increases the torque also increases, while the centrifugal tendency of the pieces A A causes the pressure, and hence their frictional grip on B B$_1$ to diminish. Slipping finally occurs between them, and any further increase in speed of the dynamo is thus checked.

These devices, however, required constant adjustment, and the use of the dynamo in this way was not long adhered to; also, with increased experience accumulators and coils for ignition purposes became so greatly improved both in capacity and reliability that the attention of designers was again directed to the battery coil system.

Multi-cylindered engines were at first fitted with a separate trembler coil for each cylinder, these receiving the necessary current impulses from the accumulators through a wipe-contact distributor. With similar coils and careful adjustment of the tremblers satisfactory running could be obtained in ordinary cases; where specially high speed and even running became necessary the difference in the lag of individual coils led to the adoption of several methods of 'synchronising' the ignition designed to eliminate this source of irregularity.

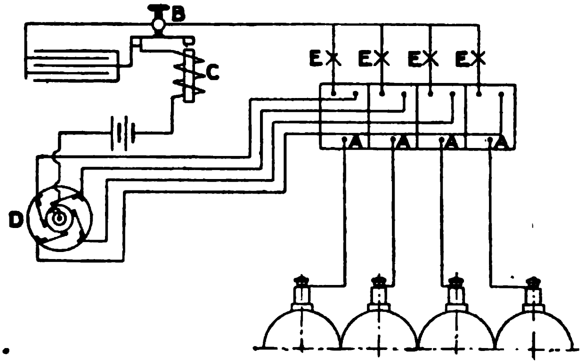


FIG. 211

Messrs. Chenard & Walcker employed a single coil and trembler for a two-cylindered engine, and connected up the two sparking-plugs in series; these accordingly sparked simultaneously under all circumstances, one only producing ignition, the other being idle; incidentally this employment of two sparking-plugs in series was claimed to give some of the igniting advantages of the oscillatory discharge.

Messrs. Wilson & Pilcher's arrangement included an induction coil for each engine cylinder with one trembler common to all; this gave a neat and convenient design and was at one time much used. The Wilson-Pilcher compound coils were usually fitted with a spare trembler as a stand-by, which could be instantly switched into action if the working trembler became defective. The wiring of a four-cylinder engine on this system is indicated in fig. 211. A, A, A, A are the four coils ; B is the trembler common to them, worked by the auxiliary coil (with primary winding only) C ; D is a four-part wipe-contact distributor ; E, E, E, E are switches enabling any coil to be cut out of the circuit at will. In the actual case the auxiliary coil and switches are all included in the coil-box with A, A, A, A.

Mr. Blake, of Kew, was one of the first to adopt a third synchronising arrangement in which only one induction coil and trembler is employed whatever the number of engine cylinders. This requires, however, a distributor for the secondary as well as for the primary current ; the insulation and general constructive details of the high-tension distributor have demanded very careful attention ; the difficulty of preserving the insulation proved considerable for a time.

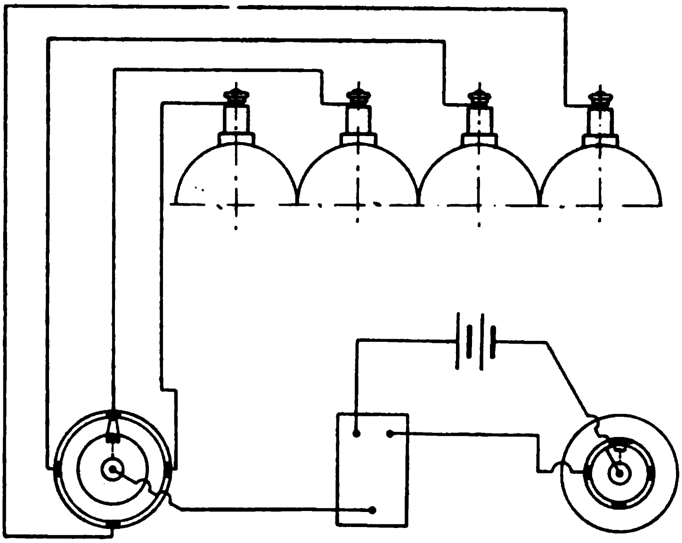


FIG. 212

Fig. 212 shows the wiring diagram for a four-cylinder engine on this system ; this is simple and self-explanatory.

Messrs. Lodge Bros. also use the single coil and high-tension distributor in their well-known ignition system.

The Lodge distributor includes a low-tension wipe-contact, and a high-tension distributor in one apparatus. The low-tension wipe-contact has been already described ; the high-tension current is carried to a spring carbon brush in light contact with a highly insulated metal ring to which a metal segment is attached ; in the course of its revolution this metal segment passes in turn other carbon brushes each of which is connected with a sparking-plug.

The next method of ignition to attract the attention of petrol engine designers was that so largely and successfully used by the makers of stationary gas and oil engines, and already described as the low-tension magneto system.

An early application to the small petrol engine was made by Messrs. Mors, who, in 1897, showed an ignition operated by a dynamo friction-driven from the fly-wheel. The dynamo current passed through an 'extra-current' bobbin and was carried inside each cylinder, where a break was produced between a movable pallet and an insulated metal rod, the pallet being worked by a cam on the half-speed shaft.

Though now superseded by the high-tension magneto system, it will be useful to preserve here also a record of the well-designed and executed low-tension magneto ignition devised by Mr. Lanchester and for long embodied in the standard engines built by the Lanchester Motor Co.

One of the engine flywheels carried two powerful bar magnets, N, S (fig. 213); this arrangement causes what are known as consequent poles to be formed, and these are located at the centre of each bar magnet. The armature about which these magnets revolve is wound with four coils, forming two independent circuits, each comprising two coils in series. One end of each of these coils is connected with 'earth,' i.e. the frame of the car. The other ends are connected to two terminals, from which leads are taken to the igniters. The armature, which is fixed relatively to the frame of the car, is concentric with, and supported by, the main shaft of the engine.

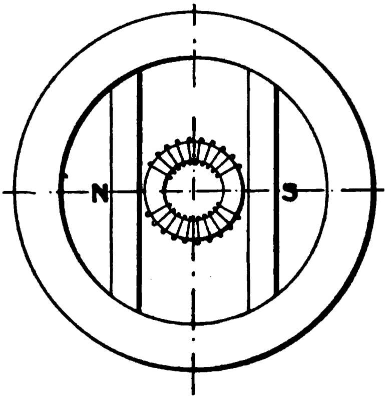


FIG. 213

The sparking-plug embodies several ingenious features, notably the method by which the intermittency of the circuit is obtained, and the ease of removal from or replacement in the cylinder.

It consists of a hollow steel plug, fig. 214, screwed internally at one end to receive a gland nut. Inside this casing is a metal bush having a fine bore, which is insulated from the casing by mica washers at either end. Through the bore of this insulated bush the 'wire' passes, making a close working fit. The end of this wire at the cylinder end of the plug has a small collar turned on it, which beds against the face of the central bush and forms a gas-tight joint when under pressure, and beyond this collar it is bent across to form a contact piece with a projection from the plug shell, so as to form a path for the current.

The other end of the ' wire ' is fitted to carry the ' ignition ' spring, which causes the make and break action when passed by the ' tweaker,' which is a projection fixed to, and revolving with, the half-speed shaft of the motor. This ignition spring is a broad, flat, thin steel plate of diamond shape. One end is gripped between insulated studs, carried on a bracket screwed to the outside of the plug casing. The broadest part is held by the ' wire,' and the remainder which forms the shortest part of the diamond is left free. It is this portion of the spring that comes into contact with the ' tweaker ' on the cam shaft as it revolves, thus causing rocking of the movable spindle or ' wire ' of the plug, and the necessary make and break of contact at the spark-gap within the cylinder.

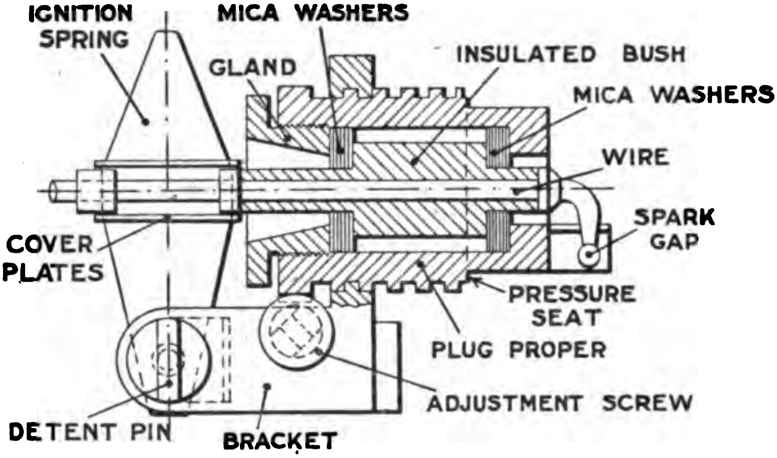


FIG. 214

The current is led from the magneto to the ignition spring by the insulated spring detents. In revolving the ' tweaker ' comes into contact with the spring, and, as revolution continues, bends it downward. This bending is distributed along the spring, and motion is communicated to the wire, the hammer end of which comes into contact with the ' anvil ' formed by the projection from the plug casing inside the cylinder, so completing the circuit ; the current then flows through this path and also through the spring and tweaker, thus forming an external circuit in addition to the internal. Thus the current from the magneto is passing through a branching circuit while the tweaker is in contact with the end of the spring. Directly, however, the tweaker overruns the end of the spring, thus releasing it, the whole of the current rushes through the internal contacts, and consequently when the 'flip back' action of the spring takes place, a heavy flaming spark results.

The advantage of the branching current lies in the fact that the external contact is always clean, and is therefore always capable of taking the whole of the current.

The internal contact may be in an electrically imperfect condition, e.g. dirty or rusty, but the whole current is shunted into it for so brief a period (between the time of internal and external break only) that but little alteration results in the 'fatness' of the spark.

As usual in low-tension magneto ignition systems when applied to petrol engines, the adjustment of the igniter required somewhat too frequent attention; the high-tension magneto system has replaced it mainly on account of its greater simplicity and convenience.

The low-tension magneto of the Albion Co. also had a fixed armature, with the bar magnets revolving round it; the armature was bolted to the casing of the engine. In general disposition this system resembled the Lanchester.

From about 1902 to 1906 low-tension magneto ignition was tried by several makers on petrol engines, but it can hardly be said to have ever become widely used; a very brief experience sufficed to show its unsuitability to the conditions of the case. In the comparatively slow running large stationary engine the low-tension magneto system is still used very extensively, and with excellent results. The parts of the sparking-plug or 'igniting block,' and the operating link-work can be made of substantial size and arranged so as to be readily accessible for adjustment and repair. In the small petrol engine, where the parts must necessarily be very lightly made, and the whole machine closely compacted, and where, moreover, the speed is from five to ten times as great as in the former class, the conditions are quite different.

The magneto machine is not essential to the low-tension system; a 4-, 6-, or 8-volt accumulator was sometimes employed in lieu of the machine, the intensity of the igniting spark at break in the cylinder being increased by inserting a laminated soft iron core about 6 ins. long by 1 in. diameter having a few turns of insulated wire upon it, in the accumulator circuit. This is essentially the ignition used in some of the Nürnberg stationary engines.

The cam and lever mechanisms operating the magneto and ignition plug proved objectionable in petrol engines, but the principal difficulties were encountered in connection with the rocking spindle and lever of the ignition plug. These were: (1) Loss of compression and consequently of power through leakage past the rocking spindle after a short period of service; (2) The rapid burning away of the contacts within the cylinder, necessitating frequent and troublesome readjustments; (3) The difficulty of maintaining satisfactory synchronisation in multi-cylindered engines; and (4) Break-down of the insulation of the fixed spindle of the plug.

While in adjustment the low-tension magneto system gave excellent results, the very hot and full spark causing prompt and regular ignition; but the constant attention needed to preserve even running at the high speeds required rendered it unsuitable for use in these small engines.

All previous devices have now (1911) been completely superseded by the high-tension magneto system, in the rapid development of which Messrs. Bosch, Eisemann, Simms, Bassée-Michel, Fuller, Lacoste, &c., have taken a prominent part. Fully illustrated and detailed accounts of the earlier types appear in the *Automotor Journal*, especially from 1904 onwards.

For a careful comparison of the high-tension magneto with the (trembling) coil-and-battery system, so far as effectiveness in firing the working mixture is concerned, the valuable work of Dr. W. Watson, F.R.S., may be referred to. Dr. Watson's results are given in detail in the *Autocar* for March 2 and November 30, 1907; after discussing the experiments, he finally concludes:

'(1) When the inlet valve is in a pocket and the plug fitted near it, neither the pressure nor the rate of burning depends at all on whether the coil or magneto be used.

'(2) If the plug be fitted actually within the combustion space, above the piston, then there is little difference so long as the mixture is nearly that giving the maximum pressure. With weak mixtures, however, there is a distinct gain in using the more intense spark of the magneto.

'(3) The well-known fact of the greater flexibility of an engine with valves in pockets than when the valves are overhead results from the presence around the plug in the former case of a mixture richer than the average of the whole, and in the latter of a mixture of average richness only.'

In the stronger mixture ignition starts more quickly and rapidly extends throughout the general mass of the charge. From some special diagrams showing the development of the explosion pressure Dr. Watson finds that the time interval elapsing between the passage of the spark and the *first appreciable rise* in pressure due to ignition is about 0·0065 second with a weak mixture, and less than one-half of this, viz. 0·003 second, when a strong mixture is used.

'(4) Where improved running has been observed, especially in multi-cylindered engines, on replacing a coil by a high-tension magneto, this is, in general, due to the improved timing obtained from the plain make-and-break of the magneto.'

Dr. Watson's earlier experiments show very clearly the inherent disadvantages of the trembler, and supply the explanation of the long well-known result that faster running and more regular firing can be obtained with a non-trembling than with a trembling coil.

With the ordinary coil and battery system it was often found that the timing of the wipe-contact was at fault, due to imperfect construction or careless adjustment of the contact blades. Irregular running arising from this cause was, no doubt, often improperly considered to be a result of faulty action in the coil.

Broadly, the advantages of the high-tension magneto over the coil and battery appear to be (1) that weaker mixtures can be fired regularly, and (2) that faster and quieter running can be obtained owing to the use of a plain make-and-break device in the magneto.

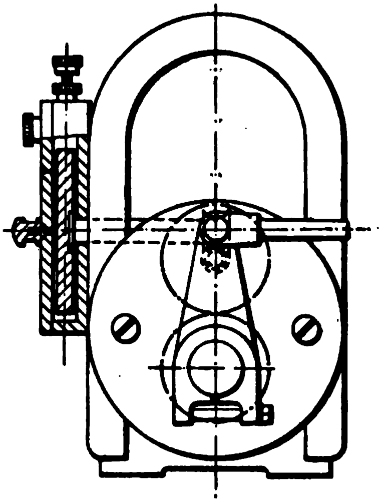


FIG. 215

The latest high-tension magneto machines exhibit in their design and execution the very perfection of arrangement and constructive skill, and are unsurpassed for effectiveness in working. Just as in the transition from tube to coil-and-battery ignition it was common to find engines fitted with both systems, the older and better-known being regarded as a reserve, so, until recently, it was usual for both high-tension magneto and coil-and-battery systems to be fitted; here the latter was mainly as a stand-by, but was also of direct service in facilitating starting. The separate coil-and-battery ignition is now, however, very generally omitted except in the larger and costlier cars, and complete reliance is placed on the high-tension magneto ignition alone; no stronger argument in support of its great reliability could be adduced.

An early high-tension magneto for ignition purposes was that patented by H. T. & H. A. Dawson, of Canterbury, No. 11720, June 28, 1900. This is fully described in the specification, and figs. 215 and 216 give an illustration of their machine. Permanent field magnets were employed with an armature of I-section, continuously rotated by the engine. Provision is made for the direction of the armature current through the make-and-break to be reversed at consecutive breaks, to ensure preservation of the platinum contacts. A coil is included in the design, the secondary current from which was delivered to the several sparking-plugs by means of a high-tension distributor, forming also part of the machine.

In the *Automotor Journal* for April 1903 a description is given of an Eisemann high-tension magneto which had then just been adopted by the Germain Motor Co., and had also been fitted to some engines of the Motor Manufacturing Co.

Much difference of opinion has existed as to the desirability of providing any means of timing the ignition when the high-tension magneto system is used, and on many engines the ignition is fixed, i.e. not capable of advancement at the will of the driver. It is true that a high-tension magneto does, to some extent, automatically time itself throughout a range of its own, owing to the increase of voltage with speed causing a corresponding decrease in the firing lag, and thus an advance in the instant of ignition. It is improbable, however, that this self-timing of the magneto will proceed *pari passu* with the advance required by the increased engine speed, and hence the provision of some timing device, either hand or mechanical, appears to be rationally defensible.

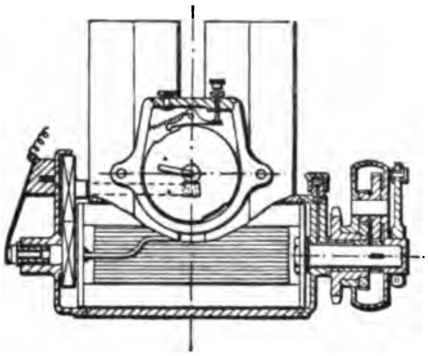


FIG. 216

The earlier Eisemann models were hand-timed by means of a spiral key fitted between the driving sleeve and the armature spindle. In the 1911 design the spiral key has been retained, but the adjustment between the driving and armature spindle is effected mechanically by a centrifugal governor contained in a case carried on an extension of the magneto base-plate. The important advantage thus gained is that the firing spark always occurs when the armature is in the maximum current position. The device is illustrated and described in detail in the *Automotor Journal* for October 29, 1910. The Bosch Co. have a similar automatically timed magneto.

In some cases where the timing is changed without angular adjustment of the armature, special extension ' horns ' are fitted to the pole-pieces of the permanent field magnets to reduce the diminution in intensity of the firing spark that otherwise occurs.

In figs. 217 and 218 are shown illustrations of one of the latest high-tension magnetos of Bosch, for four-cylinder engines of the automobile type. The rotary shuttle armature carries two windings, viz. a primary of stout wire, and a longer and finer secondary. The beginning of the primary winding is connected to the armature core which is in metallic connection with the contact-breaker disc 4. The end is connected with the block 3, which is insulated from the contact-breaker disc. The primary current is short-circuited through the platinum points 5 of the contact-breaker until the moment of break, which occurs, of course, for best effect at the instant of maximum current. The beginning of the secondary winding is connected to the end of the primary, i.e. to the insulated block 3. The end of the secondary is connected to a highly insulated 'slip' ring, 9, from which the current is taken off by an insulated carbon brush, 10, and thence carried to the rotating disc 14 of the high-tension distributor, whence it is conveyed to the several cylinder sparking-plugs by a radial contact carbon brush, 15. The secondary current returns from the engine frame to the armature core and thence, *viâ* the contact-breaker, completes its circuit. The wiring diagram for a four-cylinder engine is given in fig. 219.

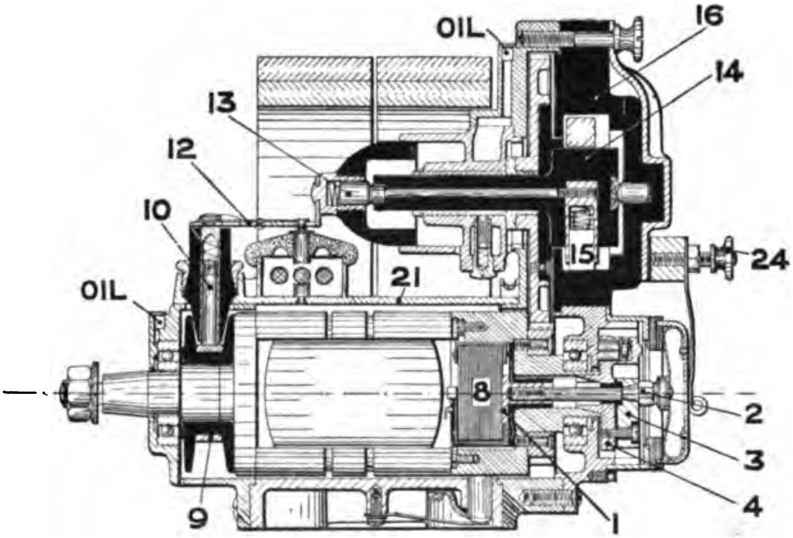


FIG. 217

Referring to figs. 217 and 218, the end of the primary winding is connected to a brass plate, 1; 2 is a screw through the centre of the plate 1, which holds the contact-breaker disc 4 in position and serves also to conduct the primary current to the block 3, carrying the platinum-pointed screw 5; 1, 2, 3, and 5 are insulated from the contact-breaker disc 4, which is in metallic connection with the armature core and thus with the *beginning* of the primary winding. Pressed against the screw 5 by the spring 6 is the platinum-pointed extremity of the bell-cranked contact-breaker lever 7, which is carried by the contact-breaker disc 4, and in thus in connection with the beginning of the primary winding. Hence it will be seen that the primary current is short-circuited so long as the platinum-pointed ends of the lever 7 and screw 5 remain in contact; the current is broken when the lever 7 is rocked; 8 is a condenser, connected, as usual, in parallel with the primary current gap.

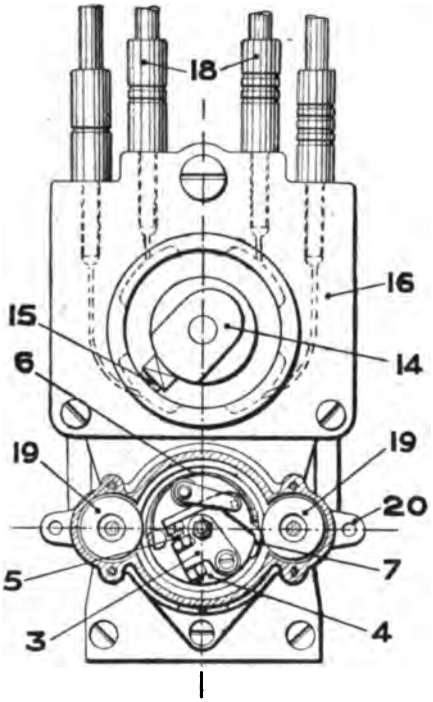


FIG. 218

Secondary Winding.—The beginning of the secondary winding of the armature is connected to the insulated system 1, 2, 3, 5, to which the end of the primary is also connected. The end of the secondary winding is carried to the insulated slip ring 9, from which a light spring

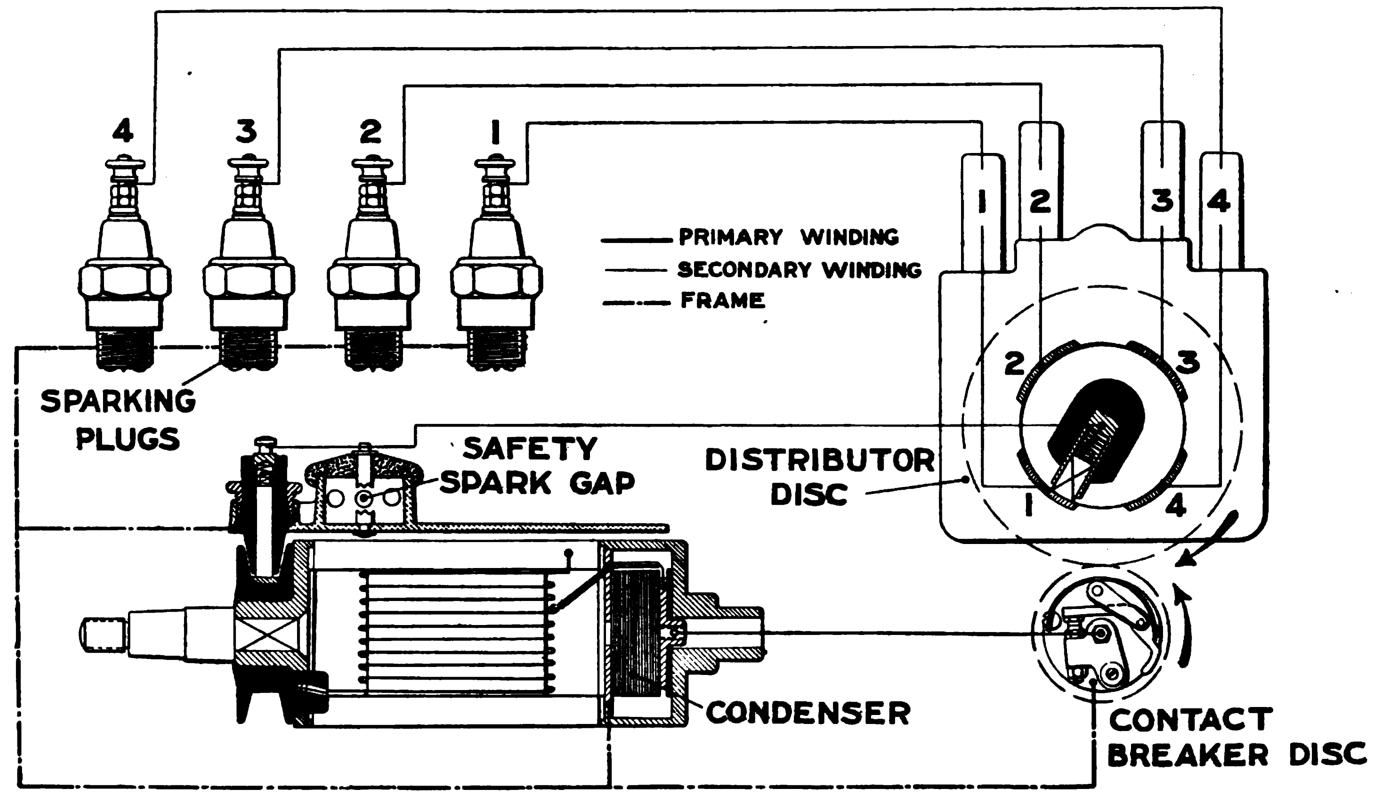

PRIMARY WINDING
SECONDARY WINDING
FRAME
SPARKING PLUGS
SAFETY SPARK GAP
DISTRIBUTOR DISC
CONDENSER
CONTACT BREAKER DISC

Fig. 219

carbon brush, 10, conducts the secondary current by way of the connecting bridge 12 to the axial carbon brush 13 of the high-tension distributor disc 14, carrying the radial carbon contact 15 to the distributor body 16, containing four contacts which are respectively connected with the four high-tension terminals 18 of the machine, as shown clearly in fig. 218. From these four terminals insulated leads carry the secondary current to the several cylinder sparking-plugs as indicated in the wiring diagram, fig. 219. For a six-cylindered engine the distributor body 16 contains, of course, six contacts, and the distributor disc 14 is suitably geared, so that the radial carbon contact 15 makes circuit with these at the proper firing instants.

With an armature of the type here used there are two current maxima per revolution, and therefore two suitable firing sparks are obtainable per revolution. Now for engines on the Otto cycle the following number of sparks is needed per revolution :

No. of cylinders . . .	1	2	3	4	6
Sparks per rev. . . .	½	1	1½	2	3

As the magneto armature gives two sparks per revolution, all that is necessary, therefore, is that this should be driven at the rates relatively to that of the crankshaft as given in the following table :

Number of cylinders.	Ratio of $\frac{\text{Armature speed.}}{\text{Crankshaft speed.}}$
1	½/2 = ¼
2	1/2 = ½
3	1½/2 = ¾
4	2/2 = 1
6	3/2 = 1½

In practice, however, for one-cylindered engines the armature is driven at *half* the crankshaft speed, i.e. at the same rate as the engine half-speed shaft ; hence a spark occurs at each revolution of the engine, alternate sparks only being effective in firing.

Similarly, in the usual arrangement of two-cylindered engines with cranks at 180°, the armature is driven at *crankshaft* speed, and here again half the sparks produced are idle ; with these engines two sparks are required in every other revolution, the intermediate revolution having no working period. In single-crank, two-cylindered engines, however, there is one working period in *every* revolution, and hence in these cases the armature is driven at *half* the crankshaft speed, i.e. at the same speed as the engine half-speed shaft, and there is then no idle spark. The important advantages gained by this practice with single- and double-cylindered engines are (1) a sufficiently high armature speed to ensure efficient firing, and (2) no separate high-tension distributor, with its attendant mechanism and fittings, is necessary, the secondary current being taken direct from the brush 10 (fig. 217) to the sparking-plug in a one-cylindered engine, or from two such brushes alternately, placed diametrically opposite one another, in the case of a two-cylindered engine, the slip ring 9 and its insulating seating being replaced in this case by an insulated disc carrying a metal segment connected with the end of the secondary winding. This segment during the armature rotation makes contact successively with these two brushes, and so causes the firing sparks to occur at the plugs.

Where a high-tension distributor is necessary it is so geared from the armature spindle as to rotate at the same rate as the half-speed shaft of the engine.

The magneto is put out of action, i.e. the ignition is 'switched off' by permanently short-circuiting the primary winding. This is effected through terminal 24 (fig. 217), which is in electrical connection with the insulated end of the primary through the system 2, 3, 5, 1; 24 is connected by wire with a switch by which it may, at will, be 'earthed' to the engine frame and thus to the armature core to which the beginning of the primary is attached. The primary circuit is thus permanently established, and the contact-breaker rendered inoperative. The width of gap at the sparking-plug should be 0·4 mm. (about 0·016") for best results. To protect the insulation of the armature and other parts of the magneto from the strains caused by excessive voltages, a safety sparking gap is fitted on the dust cover 21 (fig. 217); this is so set that the secondary current will pass across it from the connecting bridge 12 to 'earth,' i.e. the frame of the machine, and thus complete its circuit in case of breakage, or too wide a gap, in a sparking-plug. The breaking of the primary current is effected twice during each armature revolution by a pair of fibre rollers 19 (fig. 218) carried in the fixed metal cover of the contact-breaker, wiping the tail 7 of the bell-cranked lever in its passage past them. The ignition is timed by causing the break to take place earlier or later; this is effected by giving an angular turn to this cover or 'timing-plate'—and hence the fibre rollers 19—by means of a control rod attached to the boss 20; the contact plates in the high-tension distributor body 16 each extend over a sufficient arc to ensure contact being made by the radial carbon brush 15 over the whole range of timing.

As the armature position, relatively to the pole pieces, for maximum current is fixed, and the best ignition results are obtained by interrupting the primary at the instant of maximum, the alteration of the timing in this way is attended with some sacrifice of sparking efficiency. Practically a variation of about 35° about the armature spindle is found to be available, corresponding to a crankshaft variation of 35° for four-cylinder engines, and two-thirds of this, or about 24°, for six-cylindered engines. This disadvantage is removed in certain models, already referred to, wherein the armature spindle receives an automatic or hand-operated angular advance or retardation relatively to the spindle driving it, so that the interruption of the current always occurs at the moment of maximum.

Not uncommonly in high-tension magnetos the primary and secondary windings form entirely separate circuits except that both may be 'earthed' at a common terminal; it will be noted, however, from the above description of the Bosch machine, that the beginning of the secondary winding is connected to the 'live' end of the primary; thus the return of the secondary current, at least in part, is through the primary winding. In practice this is found to produce very satisfactory ignition, the firing spark being a hot, arc-like discharge capable of igniting weaker mixtures than the ordinary jump-spark.

In present practice the high-tension magneto, generally as described, is alone fitted to the engines of the smaller class of motor cars. In rather larger and more costly cars, where specially easy starting is required, and where also it is wished to obtain the advantages of 'self-starting,' the so-called 'dual high-tension system' is usually fitted.

The latest Bosch dual ignition arrangement ingeniously combines high-tension magneto and battery-coil ignition, using one sparking-plug only in each cylinder. The apparatus consists essentially of a special high-tension magneto, an induction coil with trembler and special switching arrangements, and a four-volt or six-volt battery of accumulators. The special induction coil, termed the 'starting coil' is, in cars, fitted on the dashboard with the axis usually horizontal, the switch end facing the driver and the coil case projecting forwards beneath the bonnet.

In fig. 220 the horizontal Bosch starting coil is illustrated in section and face view. The coil proper, contained in a cylinder of insulating material, is connected through the screw 5 with a bronze cover-plate, 2, carrying a wing handle, 1, by means of which it may be turned within its case so that handle 1 may be opposite either of the positions marked M (magneto on), O (off position), or A (accumulator ignition). The fixed insulating plate 6 forms the bottom of the coil *case*, and carries six metal contacts connected with terminals. A plate, 7, also of insulating material, forms the base of the coil, and turns with it by means of 1; 7 also carries 6 metal contacts, two of which are segmental. One of the contacts of both 6 and 7 is axial, and consequently these are always in electrical connection. The other contacts complete appropriate circuits dependent upon the position of the handle 1. The terminal 9 serves to attach the insulating cover 8 to the bottom of the coil, and also to connect the core

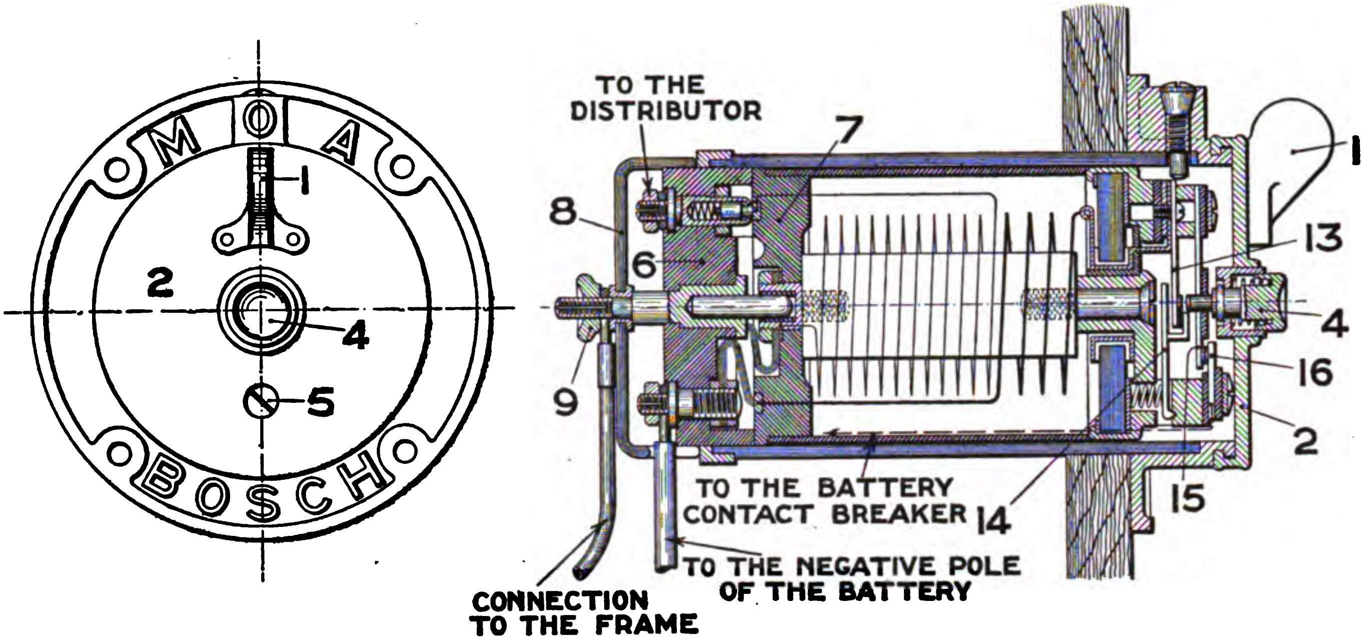

M O A
BOSCH
TO THE DISTRIBUTOR
TO THE BATTERY CONTACT BREAKER
TO THE NEGATIVE POLE OF THE BATTERY
CONNECTION TO THE FRAME

Fig. 220

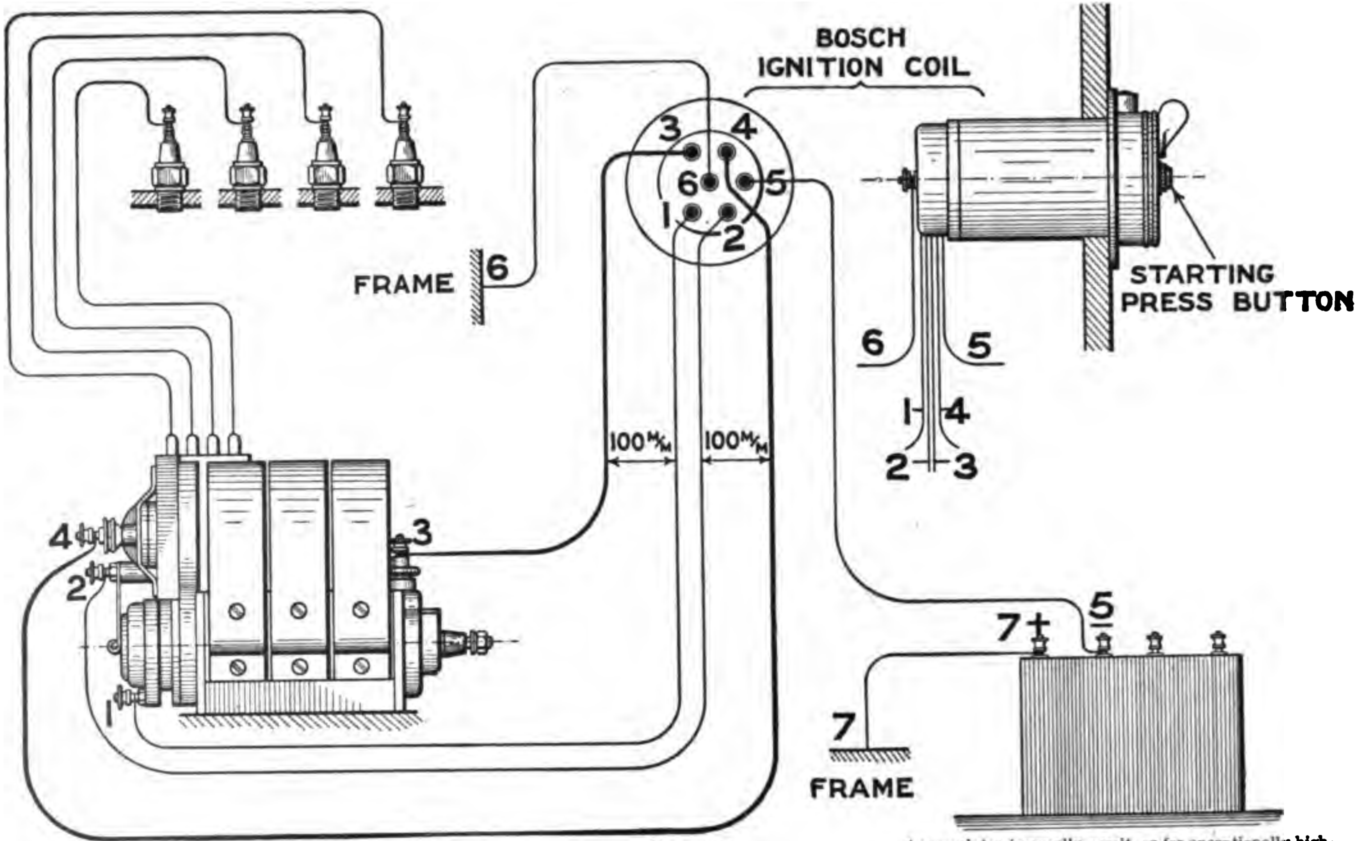
of the coil with 'earth,' i.e. the frame of the engine, through the axial contacts of 6 and 7. The beginning of the primary winding of the coil is attached to one of the segments in 7 already referred to; the end of this primary is connected with the insulated platinum-studded contact-spring 13. One end of the secondary winding of the coil is in connection, through the high-tension distributor on the magneto, with the firing plug, the return thence being, as usual, through the engine frame, to the other end.

Suppose the engine to have stopped with one piston in firing position; the switch handle 1 is moved from O to A; the push-button 4 is then gently pushed in until its platinum-tipped end makes contact with the platinum stud on 13; the primary circuit is then completed, since the positive terminal of the battery is earthed, and is thus in communication with terminal 9, and so, through the core of the coil, with the push-button 4; the negative terminal of the battery is, through terminal 5 (see wiring diagram, fig. 221), connected, *viâ* the appropriate contacts made in position A between the plates 6 and 7, with the beginning of the primary of the coil, the end of which is, as already stated, connected with 13; thus the battery current passes across the platinum contacts of 4 and 13, the coil core becomes magnetised, the trembler 13, 14 vibrates, the firing spark passes across the plug gap, and the engine starts. If the push-button 4 be now released, the switch handle remaining in the position A, in which the magneto is cut out of action, the engine will stop unless means be provided to maintain the ignition by the coil.

The means provided are as follows:

In the end view illustration of the magneto fig. 222 will be observed an insulated platinum-tipped screw, 30, and connected terminal 31, together with a platinum-studded rocking lever, 32, carried by the 'timing-plate,' 33; 30, 31, and 32 constitute the battery contact-breaker. Lever 32 is rocked by its tail being wiped by a revolving steel cam mounted behind the magneto contact-breaker disc 4, and so set that the battery and magneto contact-breakers open simultaneously. The lever 32 is earthed. Hence if the engine stops, as above assumed, in firing position, the contact 30, 32 is *open* and the battery current consequently cannot take this path, the circuit in this case being momentarily completed by aid of the push-button 4 and trembler 13, 14 of the coil, as just described.

The engine having started, however, the rotation of the magneto causes the regular operation of the battery contact breaker 30, 32; the contact 13, 4 is broken, but the battery current now passes from the positive terminal to earth, thence to 32, across the 30, 32 contact to the terminal 31; from 31 by a lead to the terminal marked 1 in the wiring diagram (fig. 221) connected to the blade 16 of the 'auxiliary



Accumulator (normally 4 volt, or for exceptionally high-speed engines, 6 volt).

Fig. 221

contact-breaker' 15, 16, in the coil (fig. 220) to be referred to later; through 16, 15, which is constantly closed during the working of the engine, to the end of the primary winding of the coil which is connected with 13; then through the primary coil and to the negative pole of the battery *viâ* an appropriate contact between plates 6 and 7 and connected terminal. Thus the battery current is complete, and the periodic breaks of the *battery* contact-breaker by the rotation of the magneto generate in the coil, now of the non-trembler type, the requisite secondary-current firing impulses which are distributed to the several plugs by the high-tension distributor of the magneto; the engine thus continues running regularly on this (non-trembling) coil ignition system.

When ignition by the high-tension magneto is desired, the switch handle 1 (fig. 220) is moved over to the position M; the coil is then cut out, the magneto primary circuit broken, and the engine continues running on the firing currents then supplied by the magneto.

The engine may stop in such a position that the battery contact breaker 30, 32 (fig. 222) is *closed*. In this case on turning the switch handle to the position A, the battery current will be short-circuited through 30, 32, and if the push-button 4 be then operated as before the trembler 13, 14 will not vibrate and the engine will fail to start.

A means is accordingly required of momentarily breaking the battery circuit through 30, 32, and this is ingeniously provided by the 'auxiliary contact breaker' 15, 16, shown in fig. 220, which, as already described, is in series with the battery contact-breaker in the battery circuit.

The procedure is now as follows:

If on pressing in the push-button 4 gently the engine fails to start through 30, 32 being closed, it should be again pushed in quickly and rather further than at first. This causes a collar on 4 to press on an insulated plate on 15 (see fig. 220), and thus break the battery circuit at 15, 16; at the same time the platinum-tipped end of 4 makes contact with the platinum stud on 13, thus causing the trembler 13, 14 to vibrate and the engine to start as before.

The coil contains the usual condenser connected in parallel with the contact breaks. It should be noted that the switch handle ought not to be left in the position A when the engine is stopped, as in the case of 30, 32 being closed, the cells would discharge themselves through this circuit in a short time with consequent damage; it is accordingly proper to bring the switch handle always to the O position when the engine is at rest.

Finally, then, with this system one has:

(*a*) A momentary ignition by trembling coil and battery.

(*b*) A permanent ignition by non-trembling coil and battery.

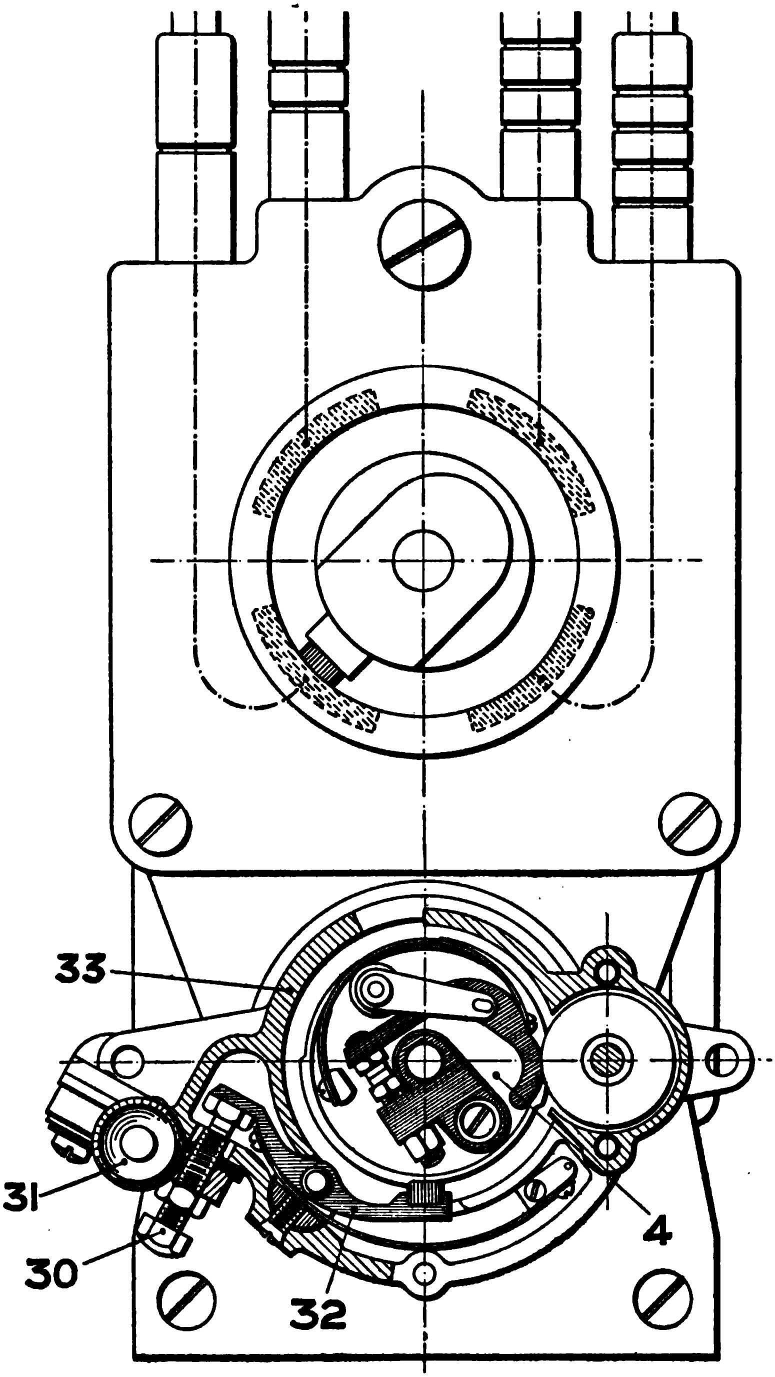


FIG. 222

(*c*) A permanent ignition by high-tension magneto.

(*d*) Only one sparking-plug to each cylinder for all cases.

(*e*) An effective self-starting arrangement.

In a later design of Bosch coil the push-button is actuated through an auxiliary handle by which it may be left permanently pressed in, with the auxiliary contact-breaker open. When this is done the engine can run, when desired, on a *trembler*-coil and battery ignition, in addition to the above advantages.

In the Bosch *double* ignition high-tension magneto two ordinary sparking-plugs are simultaneously fired in each cylinder, the secondary current course being plug-cylinder-cylinder-plug. This machine is in general features similar to those already described; broadly, the difference consists in the point that *both* ends of the armature secondary winding are normally insulated from earth, and thus two plugs per cylinder can be simultaneously operated.

The quicker rise of pressure obtained with double ignition renders less necessary the provision of timing arrangements; it is stated that with this magneto an advance of 15° is more than equal in its effect to one of 30° in ordinary cases, and also that, in general, it renders unnecessary any regulation of the timing; a further reference to this machine will be found in the concluding section of this chapter on 'Sparking-Plugs.'

The details of the machine are so worked out that the two plugs in each cylinder may be used either separately or together. Thus an equally strong spark is obtainable at slow speeds when starting the engine as with an ordinary magneto, while no increased armature winding is necessary to operate both the plugs in series when at high speed. Each set of plugs may also be instantly cut out by the switch, and thus a faulty plug may be located.

From the electrical standpoint the arrangement adopted is clearly shown in the diagram of connections, fig. 223. The ends of the secondary are coupled up to their respective highly insulated distributor brushes D^4 and D^{44}; when in full working the firing current proceeds from D^4, by way of D^5 to one plug R, thence to the frame, and from the frame to the companion plug R; back by way of D^{55} to D^{44}; thence around the secondary winding and so to D^4, completing its circuit. By means of a four-position switch either D^4 or D^{14} may, at will, be earthed, in which case the path of the firing current is restricted to one plug in each cylinder only. The other two positions of the switch give the double ignition, and magneto off, respectively.

Yet another arrangement described as the Bosch 'Dual Independent' ignition may be mentioned. Here the high-tension magneto is fitted with two separate distributors, one of which deals with the secondary current of the magneto itself as already described, the other being available to distribute the currents from a battery-coil installation. In this case the two ignitions are entirely distinct, and each serves its own set of sparking-plugs ; either or both ignitions may thus be used at will.

Messrs. Simms, regarding the dual system as mainly a means of facilitating starting, and not as necessarily providing an entirely independent ignition system in reserve, have ingeniously devised an arrangement whereby the magneto armature windings are momentarily utilised to help form an induction coil, the trembler and condenser of which are contained in a special two-way switch, usually, in cars, placed on the dashboard. The trembler, in series with the primary armature winding, is brought into action when the switch handle is turned over to the battery side for starting. In this system the magneto contact-breaker is of the *reversed* type, i.e. normally *open* instead of normally closed. A singular feature of the Simms system is that the magneto automatically takes up the firing, while the switch remains on the battery contact. At first sight it would appear as if this constituted a true dual-ignition system, but it will be observed that the engine cannot, properly speaking, be run on the battery alone, because there is no means of switching off the magneto, and the arrangement of the circuit is such that it is difficult to see how the battery could be effective if the magneto were to fail, for except in respect to the magnets themselves, and the possibility of the contact-breaker failing to close the magneto circuit at all, the battery circuit would appear to be dependent upon the maintenance of the magneto

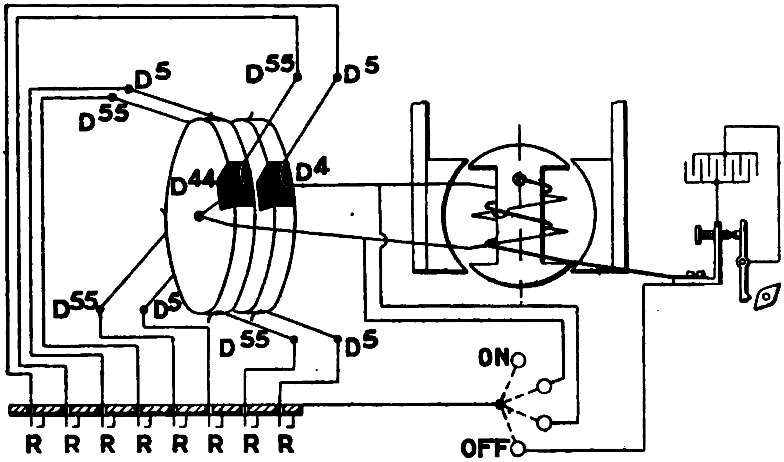


FIG. 223

circuit for its operation. The connections of the two circuits are such that the magneto is set permanently in advance of the battery circuit; this is necessary as switch starting is dependent upon late firing. An advantage of the Simms system is that the trembler maintains a chain of sparks so long as the switch is on, and this will often fire a charge where a single spark fails.

A good description of several other successful types of high-tension magneto ignition will be found in Mr. Leechman's little book on 'Systems of Electric Ignition'[1] and other works. Also in the pages of the technical automobile journals, more particularly the *Automotor* and the *Autocar*, for the past few years. The Bosch magnetos have been selected as typical, and dealt with in some detail in order to render clear the essentials of this latest and most successful system of electric ignition, and to illustrate the high degree of designing and constructive skill that has been brought to bear upon the manufacture of these very serviceable little machines in recent years.

SECTION III

SPARKING-PLUGS

A most important item in all high-tension electric ignition systems is the sparking-plug, and in the early days this detail was a source of frequent breakdown. In the non-compressing Lenoir engine, as already mentioned, the difficulties experienced, largely in connection with the plugs, did much to restrict the use of the engine commercially. With the adoption of compression the difficulties increased, owing to the higher temperatures attained and to the increased resistance of the spark-gap. Even now, if the compression in a petrol engine exceeds about 120 lbs. per sq. in., trouble usually arises with the sparking-plugs, though largely through the electrodes becoming incandescent and then causing pre-ignition. In the early sparking-plugs two insulated electrodes were usually fitted; fig. 191, *ante*, shows the Lenoir plug in section, the two wire electrodes being both embedded in an insulating matrix carried in a metal casing.

The accompanying illustration, fig. 224, shows also an early plug of similar type by Daimler; the plug is fitted in a pocket communicating with the combustion chamber, part of the intention of this arrangement being to prevent the formation of sooty and oily deposits on the inner end of the insulator containing the electrodes. On the passage of the spark the mixture in the plug pocket ignites almost instantaneously, and a tongue of flame is thence projected into the combustion chamber, thus causing a prompt ignition of the whole working charge. This mode of locating the plug has long been adhered to by some engine builders, notably Messrs. de Dion and Messrs. Lanchester.

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[1] The Car Illustrated, Ltd., London. 1907.

M. Charles Benz's first motor, constructed in 1878, was fitted with battery and coil ignition ; the Benz engines used a sparking-plug with porcelain-insulated platinum wire electrodes. The special sparking-plug used in the early stationary Benz engines is shown in fig. 197.

In all modern cases—excepting only the 'double-pole' design used when two or more sparking-plugs are connected up in series and fired simultaneously—there is a single central insulated stalk, spindle, or electrode, passing axially through an insulator of porcelain, mica, or steatite, the return circuit being through the frame of the engine itself.

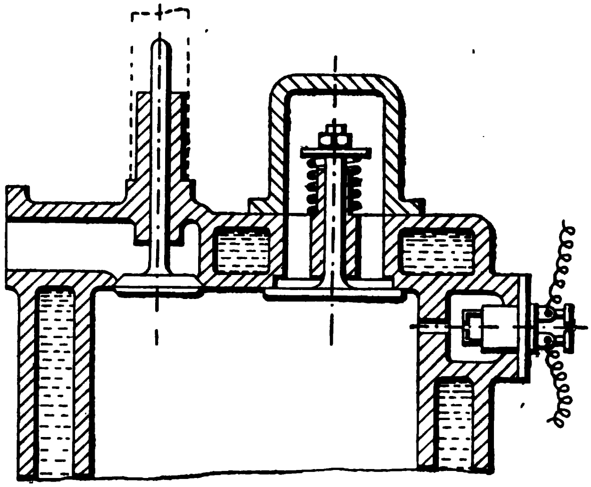


FIG. 224

A great stimulus to the production of a satisfactory type of sparking-plug was provided by the small air-cooled, very fast-running petrol engines built by Messrs. de Dion during the period 1896–1900 for driving their motor tricycles ; these were fitted with electric ignition, a battery, non-trembling coil, and plug being used.

Messrs. de Dion soon acquired a very extended and valuable experience, and produced designs of plugs that long enjoyed an unique reputation. Even at the present time the ignition bosses on petrol engine cylinders are usually bored and tapped to the de Dion standard 60° thread—viz. 18·2 mm. diameter over threads, with a pitch of 1·5 mm.—a striking testimony to the widespread use of their plugs.

Early failures were due: (1) to the electrodes being made of too small wire, or of unsuitable material, and so becoming over-heated, or burning away at the gap so rapidly that frequent readjustment became necessary; (2) to unsuitable insulating material, which in some cases disintegrated with the heat after short service, or absorbed oil and lost its insulating properties; (3) to breakage of the insulating material due to imperfections of design, proper provision not being made to allow for the expansion of the several parts of the plug when heated; (4) to failure of the spark due to the design favouring the accumulation of soot or dirty oil on the inner end of the insulator.

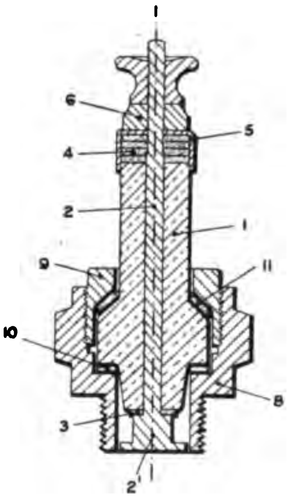


Fig. 225

All these sources of failure have now been practically eliminated by experience, and the latest types of plug are exceedingly durable and satisfactory. The central stalk, formerly frequently a fine platinum wire, is now a stout rod, very commonly of pure nickel. The insulating material is of specially manufactured porcelain, or of mica composed of a large number of compressed and consolidated discs, or of steatite; glass has occasionally been used also. The expansion of the several parts of the plug is adequately provided for, and the insulator usually terminates some little distance back from the spark gap, thus preventing failure of the ignition through accumulations of oil or soot.

Fig. 225 shows in section the de Dion 1911-type standard sparking-plug for use with the now almost universal high-tension magneto ignition.

The insulated electrode 2 is a stout metal rod carried through the porcelain insulator 1 and terminating in a four-pointed 'star' disc 2^1, between which and the plug body 8 the igniting sparks occur. Leakage between the electrode and porcelain is prevented by a small asbestos washer 3, and the difference of expansion between the electrode and porcelain is taken up by the asbestos washers 4, covered by a brass cap 5. The electrode, porcelain, and washers are drawn up together by the brass hexagonal nut 6. The gland nut 9 bears upon the coned surface of the porcelain as shown, and thus centres it in the body when screwed up; 10 and 11 are copper and asbestos washers to ensure gas-tightness. It will be seen that all parts are of substantial proportions, and that no cement or wires are used.

Fig. 226 shows the Lodge single-pole plug in section, together with a plan showing the three spark gaps. The insulated electrode is here of pure nickel, as also are the three electrodes attached to the plug body.

It will be noted that the porcelain insulator terminates at some distance from the sparking points. The means by which the expansion of the plug parts is provided for while retaining gas-tightness everywhere is apparent from inspection of the section.

This plug is equally suitable for use with the Lodge coil or the high-tension magneto system; the spark occurs well within the combustion chamber; the sleeve of the porcelain prevents leakage of the high-tension discharge over the insulating surface; this sleeve keeps at a sufficiently high temperature to prevent carbon deposit, while not so high as to cause any risk of pre-ignition. For use with the Lodge coil, the spark gaps are each 0·02″ in width.

Fig. 227 shows in section and face view a large gas engine sparking-plug designed by Messrs. Lodge Bros.; the body of the plug is of considerable length to bridge across the water-jacket of the cylinder; the insulation is of mica; the electrodes are of massive design in pure nickel; three spark gaps are provided, each about 0·03″ wide.

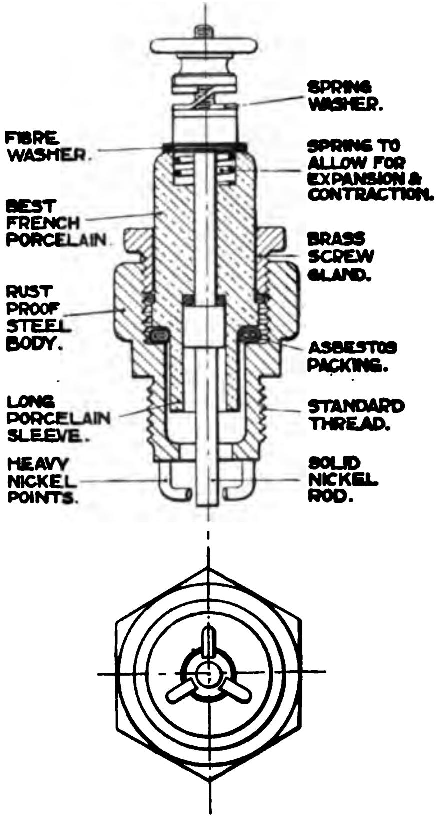


Fig. 226

In fig. 228 another recent Lodge plug for large stationary engines is shown; here also the insulation is of mica, but the nickel electrodes are so arranged that the width of spark gap can be very nicely adjusted by screwing the nickel ring inwards or outwards and thus causing its inner circular edge to approach or recede from the conical nickel bolt head fixed to the central insulated stalk. The spark gap is here a narrow annulus about 0·03″ wide. The general structural details are rendered clear by the notes accompanying the illustration.

The E.I.C. sparking-plug, 1911 type, is illustrated in fig. 229. In this plug the insulating material is of steatite, or ‘soapstone’; the insulator is conical in form within the body of the plug, and so fitted that the explosion tends to press it closely to its seat, thus preserving gas-tightness; the design renders unnecessary the usual screwed gland and packing washers. The electrodes are of pure nickel.

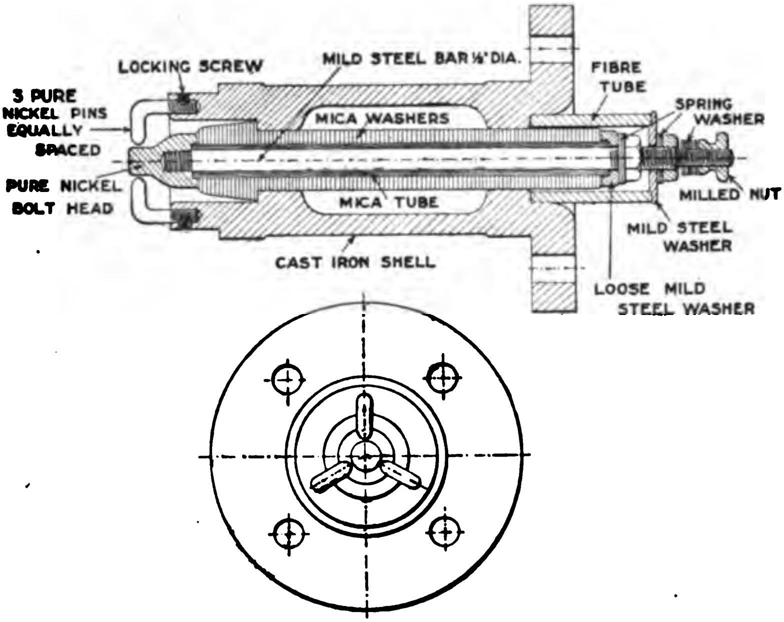


FIG. 227

The ‘New Bosch’ three-point sparking-plug is shown in half section in fig. 230, and an enlarged view of the electrodes is given in fig. 231.

These plugs are made with one, two, three, and four sparking points; the ordinary single-point plug without knife-edged electrode is recommended for use in high-speed, high-compression racing engines; the two- three- and four-point plugs with knife-edged electrodes are considered better for the engines of normal touring cars, especially at low speeds. In the makers' view the advantage of multiple sparking points is that the plugs last longer without adjustment; when one pair of points burn away the firing spark selects another pair, and thus the working life of the plug is lengthened in direct proportion to the number of body electrodes, i.e. of 'sparking points.' The characteristic knife-edged or flattened body electrodes are clearly shown in fig. 231 ; this construction aims at the production of a short ribbon of spark with the object of getting regular and crisp firing.

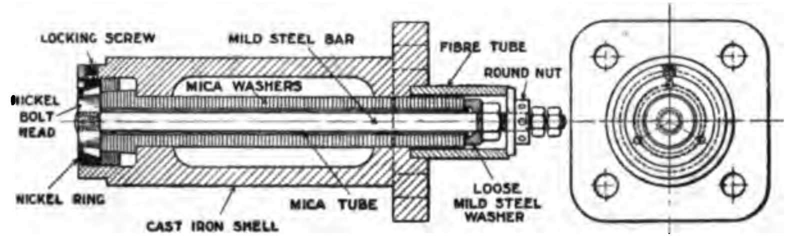


FIG. 228

In the three-point plug of fig. 230 the stout central electrode of nickel is screwed tightly into a steatite insulator ; it will be noted also that this insulator is coned away from the electrode at the firing end in order to prevent failure of ignition by shorting through oily or sooty deposit.

For large stationary gas engines using high-tension magneto ignition Messrs. Bosch have recently issued the specially designed heavy plug shown in fig. 232 ; this requires a 1" gas thread in the plug boss on cylinder. The plug is of substantial proportions ; the central insulated electrode is a stout nickel rod partially shielded at its inner end by the two domed body-electrodes A A as shown, thus preventing failure of the ignition through deposition of oily or sooty matter. The insulating material used is steatite.

In the recently introduced 'Mascot' plug a vitreous enamel of special composition is used to ensure gas tightness between the insulator and the metal body of the plug.

A section is shown in the accompanying fig. 233 ; the details of construction are rendered clear by the descriptive notes. With the ordinary plug the asbestos-packed gland has to be screwed up hard in order to obtain gas-tightness, and breakage of the porcelain insulator sometimes occurs as a consequence ; it is said that the special enamel of the Mascot plug preserves an absolutely tight joint without bringing any excessive strain upon the insulator. This plug is becoming largely used.

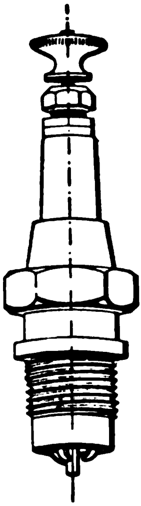


FIG. 229

Claims for improved ignition are made by makers of plugs having sparking ends of special form. In the 'Cup' detonating plug, for example, the central insulated electrode terminates in an expanded portion containing a pair of nearly hemispherical depressions (fig. 234); the two round-pointed ends of the body electrode terminate at

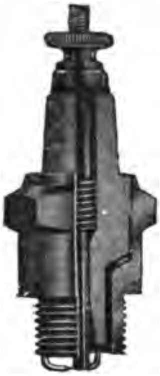


FIG. 230

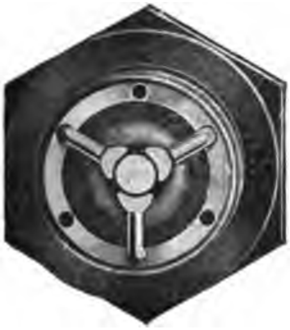


FIG. 231

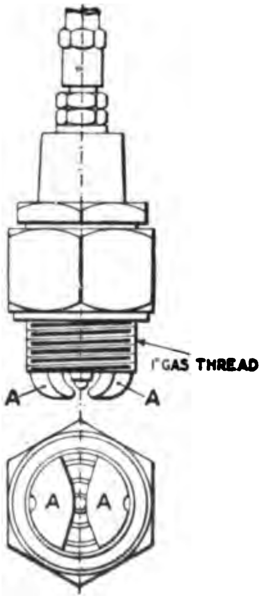


FIG. 232

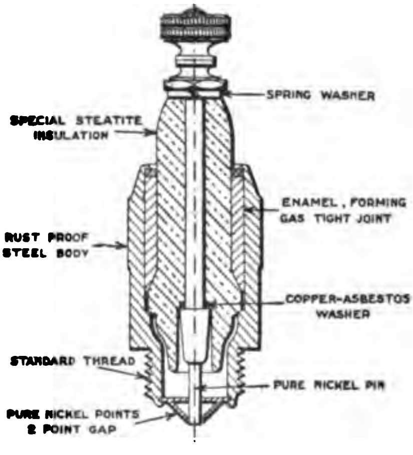

SPRING WASHER
SPECIAL STEATITE INSULATION
ENAMEL, FORMING GAS TIGHT JOINT
RUST PROOF STEEL BODY
COPPER-ASBESTOS WASHER
STANDARD THREAD
PURE NICKEL PIN
PURE NICKEL POINTS 2 POINT GAP

FIG. 233

about the centre, respectively, of each hemispherical surface. The electrodes are here of nickel and the insulator of porcelain. The cup-shaped depressions have a radius of about 1¼ millimetres; these plugs have attracted some attention, and an interesting series of experiments has been carried out by Dr. A. M. Low, whose results are fully illustrated and described in *Motor Cycling* for January 2, 1912.

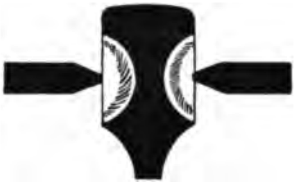


FIG. 234

On the other hand, in the 'Samson' plug manufactured by the Electric Ignition Co. the electrodes terminate in small nickel spheres, as shown in the accompanying fig. 235. The discharge takes an arched course between the balls, and it is claimed that the massive proportions of the electrodes prevent the occurrence of pre-ignition.

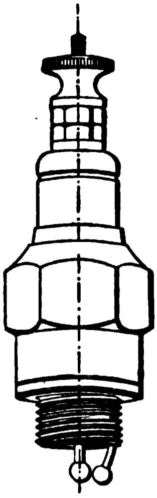


FIG. 235

Double Ignition.—It has long been known by drivers of motor cars that the running of a petrol engine is much affected by the quality of the igniting spark, and two ignitions to each cylinder have been occasionally used for some years. In certain racing engines as many as four ignitions per cylinder have been employed. A somewhat common practice from about 1905 to 1910 was to fit both coil-and-battery and high-tension magneto ignition independently, the former being intended mainly as a stand-by and also to facilitate starting. Many drivers used both ignitions simultaneously, and where properly fitted the performance of the engine was very sensibly improved thereby. In large gas engines, as already described, the benefit of double ignition has long been recognised. Prof. W. Watson's experiments[1] on a four-cylinder 85 × 120 Clement-Talbot petrol engine using high-tension magneto double ignition, the sparking-plugs being in series (fig. 236), showed, as was anticipated, that the power developed was materially increased by the use of two ignition plugs, the advantage gained increasing with the speed; his results are as follows:

r.p.m. of engine	1100	1600
HP, single ignition	18·4	26·0
HP, double ignition	20·8	29·2

---

[1] *Proc. Inst. A.E.*, 1909.

This is due to the more rapid ignition obtained with two sparks, which becomes of greater importance at increased speed. The plugs were placed over the inlet and exhaust valves respectively, these being contained in pockets on opposite sides of the cylinder.

In fig. 237 are shown diagrams obtained by Dr. Watson with his mirror-diaphragm indicator, illustrating very clearly the advantage resulting from double ignition. In each case the instant at which the spark passed is indicated by a × ; the upper diagram was obtained with single ignition, the lower with double. The engine speed was 1600 r.p.m. It will be noted that with double ignition the maximum pressure was attained in $\tau_2 = 0{\cdot}0037$ second after the occurrence of the double spark, while when single ignition only was employed the time interval increased to $\tau_1 = 0{\cdot}0055$ second. Thus in this instance the use of two sparking-plugs reduced the time of rise of pressure by about one-third. The lower diagram shows also a considerably higher explosion pressure.

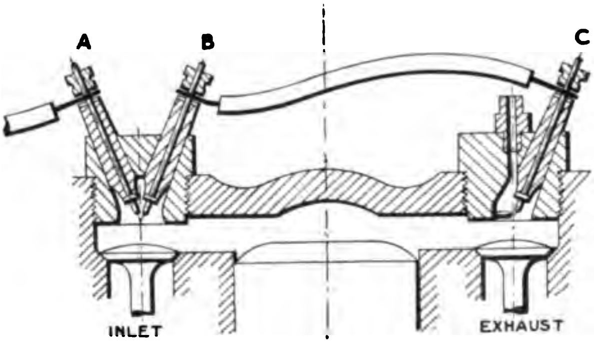


FIG. 236

The plug over the exhaust valve is not, in general, so effective as that over the inlet, owing to the lesser likelihood of a rich ignitible mixture surrounding it at the instant of firing ; in almost all cases the principal ignition is placed as near the inlet valve as possible for this reason. Several cases have been observed by the authors in which the second or stand-by system alone was incapable of causing ' crisp ' and regular ignition, due to the plug not being located advantageously. From this cause also differing degrees of improvement resulting from the use of double ignition are observed in different engines.

When double ignition is regularly used, in order to ensure that the sparks shall occur simultaneously at the two plugs, it is necessary that these be connected up in series. Owing to the almost universal practice of ' earthing ' one end of the secondary circuit to the engine, one of the sparking-plugs must, in general, be a 'double-pole' plug, containing an insulated electrode to carry the secondary current on to the second plug before its return, *viâ* the engine frame, to the magneto. The general adoption of the 'de Dion' plug thread as the

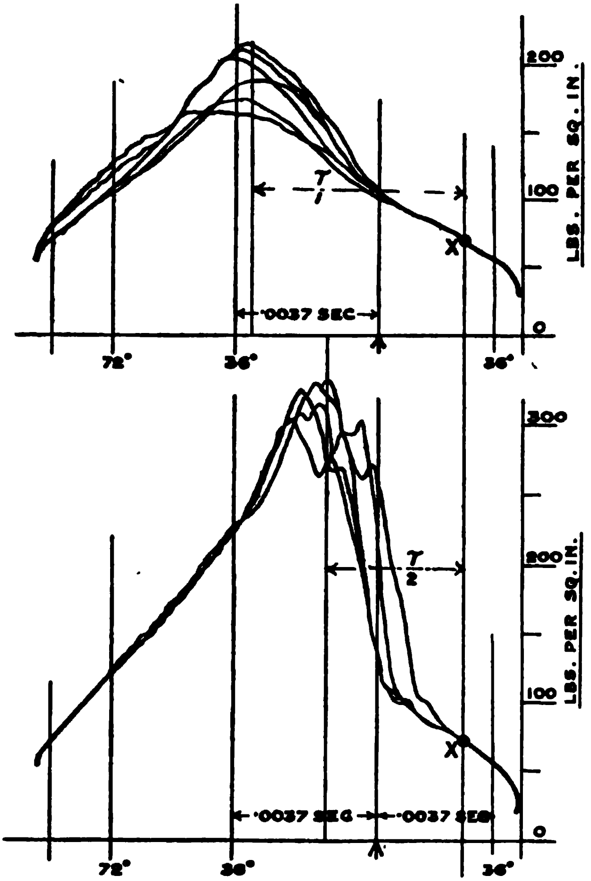


FIG. 237

standard for motor car engines has made the design of a satisfactory double-pole plug with standard thread a problem of considerable difficulty, and for some time special fittings were used; fig. 236, from Prof. Watson's Paper, shows one arrangement adopted; A and B are special sparking-plugs having a single central insulated electrode only, and carried in a screwed metal holder as shown; C may be an ordinary sparking-plug. The secondary current enters at A, sparks across within the cylinder, between A and B, is conducted from B to C by the lead shown, sparks across the gap of C, within the cylinder, to the engine itself, and is thence returned to the magneto.

Recently, however, it has been found possible to construct double-pole plugs within the narrow limits imposed by the standard plug thread, and as a typical case a section is shown in fig. 238 of the double-pole plug patented by Messrs. Lodge Bros. The details of construction are sufficiently clear from the figure and accompanying notes. The secondary current lead from the coil or magneto is connected to the terminal A, and after passing across the spark gap is conducted by a lead connected to the terminal B to the second ordinary sparking-plug. It will be seen that in this plug the spark occurs well within the combustion chamber, the ‘ reach ’ in the design shown being one inch. When used with the Lodge coil, the spark gaps in this, and the connected single-pole plug, should be each between 0·3 (0·012 in.) and 0·4 (0·016 in.) of a millimetre in width. As the electrodes are both insulated, and necessarily project some way into

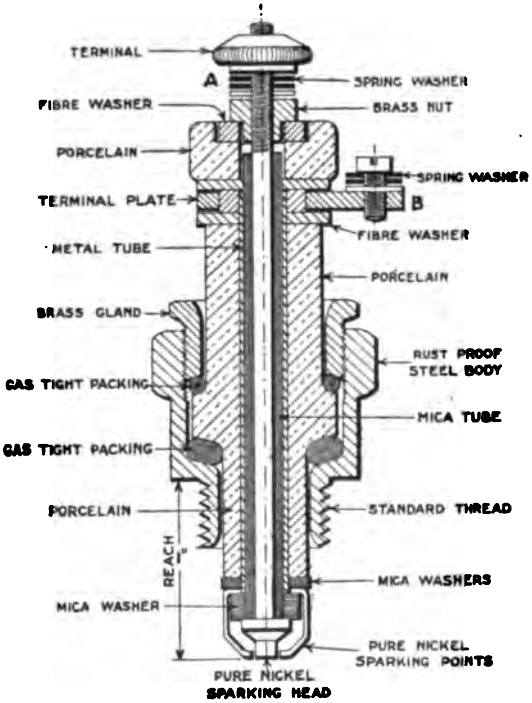


FIG. 238

the cylinder, the heat cannot so readily escape from them as in an ordinary plug, and accordingly it is probable that if pre-ignition should occur, it will originate with the double-pole plug in the cylinder.

In double ignition obtained by the use of the Bosch double-ignition high-tension magneto, ordinary single-pole plugs only are necessary, as in this machine, contrary to usual practice, *both* ends of the secondary winding are carried to insulated terminals, those being directly connected respectively to the two plugs. The secondary current sparks across from the central electrode of one plug to the cylinder casting, and simultaneously from the cylinder casting to the central electrode of the other plug.

This magneto is so arranged that either or both ignitions may be switched on at will; the authors have witnessed cases in which a marked increase occurred in the speed of a loaded engine merely on switching in the second ignition. In two tests the following results were observed on changing from single to double firing, no other alteration whatever being made:

	r.p.m.	r.p.m.
With single ignition . .	950 . .	950
,, double ,, . .	1100 . .	1150

As the power was absorbed by a fan dynamometer the output of power varied as the cube of the speed, and the increase was thus marked in this instance. In the small petrol engines of cars the difficulty of finding an effective position for the second plug very frequently results in no perceptible gain being obtained by the adoption of double ignition in consequence of the additional plug being partly or wholly inoperative.

CHAPTER IV

SPEED REGULATION, GOVERNORS, AND GOVERNING METHODS OF INTERNAL COMBUSTION ENGINES

ALL internal combustion prime movers are of the reciprocating type, and very many ' stationary ' engines are of the single-cylinder, single-acting, horizontal pattern working at present almost universally on the ' Otto ' cycle.

In general, prime movers are required to maintain a considerable degree of regularity in their speed under all circumstances ; the single-cylinder, single-acting Otto cycle engine is very unfavourably circumstanced in the production of uniform rotatory motion, since a working impulse occurs only, at most, once in every four strokes ; the momentum of the several moving parts of the engine has thus to maintain the motion during at least three-fourths of the time.

In many applications the external load on the engine is practically constant, and it is necessary that the revolution speed of the engine shall then also vary between narrow limits only ; these limits differ according to the nature of the work to be done by the engine ; in some cases as, e.g. where the engine drives a pumping plant, the load is subject to large periodic fluctuations. The method of dealing with such cases does not, however, differ essentially from that where the load is constant, and accordingly in this chapter the treatment of the simpler case of a constant external load only will be described in detail.

It is the function of the flywheel to diminish the ' cyclic ' variations of speed arising mainly from the inequality between the working impulses on the piston, or pistons, and the external load resistance whether constant or otherwise, and this diminution of irregularity in speed is greater as the flywheel is made larger and heavier.

The excess energy developed during the working impulse is stored up in the rim of a heavy flywheel, and returned therefrom during the non-working strokes, violent fluctuations of speed being thus prevented; the flywheel rim is accordingly an energy reservoir, the content of which attains a maximum and minimum value once in each complete cycle of the engine. Permanent alterations in the external load would result in permanent alterations in the engine speed. To prevent this latter consequence is the special function of the governor, and will be dealt with later in this chapter.

It is thus of practical importance to have a method of determining suitable flywheel dimensions for a given engine in order that the cyclic fluctuations of speed in normal load running shall be within an assigned limit ; such a method we now proceed to describe in some detail, and in order to assist the reader it has been thought best to work out an actual concrete case.

Accordingly we select that of a single-cylinder, single-acting, ' Otto ' cycle, horizontal gas engine of 12 in. bore and 21 in. stroke, running at a speed of 200 revolutions per minute—the corresponding piston speed being 700 ft. per minute ; the connecting-rod is five cranks in length, and the mass of the parts assumed as reciprocating is taken at 4 lbs. per sq. in. of the area of the piston. Following the usual practice, this mass includes the piston and adjuncts, and, roundly, half the weight of the connecting-rod. The value of the reciprocating mass has been taken somewhat high intentionally in order that the effect of reciprocating inertia may be the more easily discerned in the diagrams given later ; it is a value which is attained in large gas engines, though it should be noted that in the small high-speed motors of the automobile type the figure is reduced to between 0·4 and 0·6 lb. per sq. in. of piston area.

It is first necessary to describe the graphical process whereby the tangential effort at the crank-pin is obtained from the pressure acting at the crosshead. In Fig. 239, let O P, P C, represent the crank and connecting-rod of an engine, in any configuration ; producing O P to meet a perpendicular through C to the line of stroke determines I, the

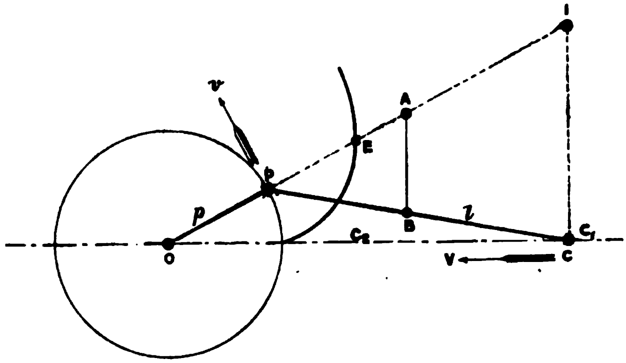


FIG. 239

instantaneous centre of rotation of the connecting-rod; hence, if v, V, respectively denote the velocities of P and C, we have:

$$\frac{V}{v} = \frac{IC}{IP} \tag{1}$$

But the rate of working at C and P is the same; hence, if F denote the force at C in the direction CO, and f that at P in the direction of v, i.e. if f be the tangential effort at the crank-pin, we have $FV = fv$, whence:

$$\frac{f}{F} = \frac{V}{v} \tag{2}$$

Consequently, from (1):

$$\frac{f}{F} = \frac{IC}{IP} \tag{3}$$

Now, as will be seen directly, F may be easily determined in any given case; to the assumed scale, then, set off PA to represent F, and draw AB vertically to meet the connecting-rod, produced, if necessary, in B; then it is obvious that $\frac{AB}{AP} = \frac{IC}{IP}$, and accordingly by (3):

$$\frac{f}{F} = \frac{AB}{AP} \tag{4}$$

But, by the construction, AP measures F; hence, to the same scale, $AB = f$. Lastly, set off from P, $PE = AB$; then E is a point on the 'polar curve of tangential effort at the crank-pin.'

By repeating this simple construction for a number of different positions of the system further points are obtained, and the curve may then be traced through them.

Determination of F.—Referring to fig. 240, DQRS is an actual normal indicator diagram from a 'National' gas engine, plotted upon the line of stroke C_1C_2 as base, a convenient scale of pressures being assumed—in this case 0·02 in. = 1 lb. per sq. in. The force requisite to produce the actual acceleration of the reciprocating parts for every position of the crosshead C must next be ascertained; this is derived from the acceleration curve drawn by Prof. Klein's method as described in Chapter VII and Appendix I; it is well, however, to calculate the initial ordinate, viz. at C_1; thus we have (*v.* pp. 509–10):

$$\text{Acceleration at } C_1 = \omega^2\rho\left(1 + \frac{\rho}{l}\right)\frac{\text{ft.}}{\text{sec.}^2} \tag{5}$$

where ω is the angular velocity of the crank-pin in radians per second, corresponding to n revolutions per minute, and thus is equal to $\frac{2\pi n}{60}$. Here $n = 200$, whence $\omega = 20{\cdot}9$ radians per second. Also $\rho = \frac{10{\cdot}5}{12}$ ft., and $\frac{\rho}{l} = \frac{1}{5}$; hence by (5) :

$$\text{Acceleration at } C_1 = 458{\cdot}5 \ \frac{\text{ft.}}{\text{sec.}^2}$$

The mass accelerated is, by supposition in this case, 4 lbs. per sq. in. of piston area ; the necessary force in lbs. weight per sq. in. of piston area is accordingly :

$$\frac{4}{g} \times 458{\cdot}5 = 57 \text{ lbs. per sq. in.}$$

That is, when the piston is at the end C_1 of its stroke, and the engine is running at 200 r.p.m., a pressure of 57 lbs. per sq. in. is necessary to produce the acceleration of the reciprocating parts, the excess above this, if any, alone remaining to produce driving effort at the crank-pin. Lay off, therefore, $C_1T = 57$ lbs. per sq. in. to the pressure scale of the diagram in fig. 240, and then complete the curve of accelerating force TXVU by aid of Klein's construction. The polar curve of tangential effort at the crank-pin may now be obtained by aid of the method shown in fig. 239 ; thus when the crosshead is at C in the working stroke, the whole forward pressure on the piston is CQ, but of this CX is absorbed in accelerating the reciprocating parts, leaving, therefore, the remainder, XQ, to produce positive tangential effort at the crank-pin. Set off, then, PA along OP produced, equal to XQ ; drop AB vertically to meet PC in B ; then AB measured to a scale of 0·02 in. = 1 lb. per sq. in. gives the tangential effort at the crank-pin in the position shown, per sq. in. of area of the piston ; take PE equal to AB, then E is a point on the polar curve of crank-pin effort ; other points are next similarly obtained, and the curve finally traced through them.

It will be observed that from C_1 to V the reciprocating parts are increasing in velocity, attaining their maximum speed at V ; from V to C_2 this velocity is destroyed ; hence from C_1 to V part of the piston pressure is absorbed by the reciprocating parts, but this is returned during the retardation period VC_2. The areas C_1VT and C_2VU are equal and of opposite sign, no work, of course, being gained or lost through inertia of parts ; the effect, however, is to diminish the crank-pin effort during the first portion of the working stroke, and increase it during the latter part, thus tending to render this more uniform than would otherwise be the case.

For the return (i.e. exhaust) stroke the vertical intercepts between

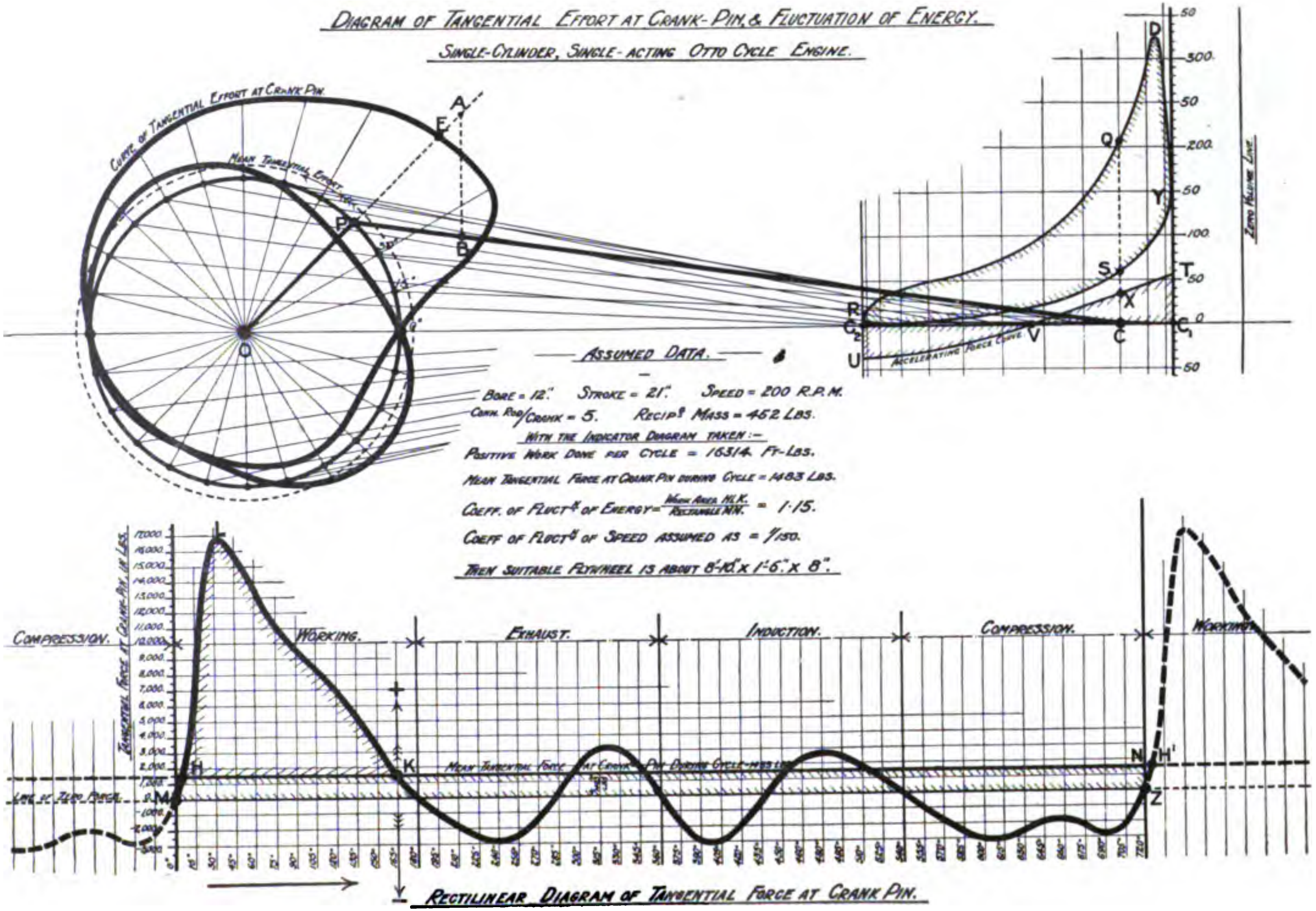


DIAGRAM OF TANGENTIAL EFFORT AT CRANK-PIN & FLUCTUATION OF ENERGY.
SINGLE-CYLINDER, SINGLE-ACTING OTTO CYCLE ENGINE.
CURVE OF TANGENTIAL EFFORT AT CRANK PIN
MEAN TANGENTIAL EFFORT
A
E
B
P
O
D
Q
Y
S
T
X
R
C2
U
V
C
C1
ACCELERATING FORCE CURVE
ZERO VOLUME LINE
50
300
50
200
50
100
50
0
50
ASSUMED DATA.
BORE = 12". STROKE = 21". SPEED = 200 R.P.M.
CONN. ROD/CRANK = 5. RECIPG MASS = 452 LBS.
WITH THE INDICATOR DIAGRAM TAKEN :-
POSITIVE WORK DONE PER CYCLE = 16314 FT-LBS.
MEAN TANGENTIAL FORCE AT CRANK PIN DURING CYCLE = 1483 LBS.
COEFF. OF FLUCTN OF ENERGY = WORK AREA HLK. / RECTANGLE MN. = 1·15.
COEFF OF FLUCTN OF SPEED ASSUMED AS = 1/150.
THEN SUITABLE FLYWHEEL IS ABOUT 8'-10" X 1'-6" X 8".
TANGENTIAL FORCE AT CRANK-PIN, IN LBS.
17,000
16,000
15,000
14,000
13,000
12,000
11,000
10,000
9,000
8,000
7,000
6,000
5,000
4,000
3,000
2,000
1,000
0
-1,000
-2,000
-3,000
COMPRESSION.
WORKING.
EXHAUST.
INDUCTION.
COMPRESSION.
WORKING.
LINE OF ZERO FORCE
MEAN TANGENTIAL FORCE AT CRANK PIN DURING CYCLE = 1483 LBS.
L
H
M
K
N
H'
Z
+
−
0° 15° 30° 45° 60° 75° 90° 105° 120° 135° 150° 165° 180° 195° 210° 225° 240° 255° 270° 285° 300° 315° 330° 345° 360° 375° 390° 405° 420° 435° 450° 465° 480° 495° 510° 525° 540° 555° 570° 585° 600° 615° 630° 645° 660° 675° 690° 705° 720°
RECTILINEAR DIAGRAM OF TANGENTIAL FORCE AT CRANK PIN.

FIG. 240

the accelerating force curve and the exhaust line of the diagram must be used; a little consideration will show that the crank-pin effort is negative (i.e. the crank-pin drives the reciprocating parts) during the period C_2V and positive during VC_1.

During the next outward (i.e. induction stroke), the intercepts to be taken are between TVU and the induction line of the indicator diagram; and lastly during the compression stroke, between TVU and the compression curve C_2SY; it will be seen that the crank-pin effort is negative throughout the compression stroke.

The polar curve of crank-pin effort is not directly used, but from it is immediately derived the figure shown in the lower part of fig. 240 entitled 'Rectilinear Diagram of Tangential Force at Crank Pin.' This is obtained by conceiving the circular path of the crank-pin during a complete cycle (i.e. two revolutions in this case) to be laid out along a straight line MZ as base, ordinates being drawn at the several points corresponding to the crank-pin positions on the polar curve; along these ordinates are laid off, to some convenient scale, lengths representing the crank-pin effort taken from the corresponding point of the polar curve. The advantage of this mode of exhibiting the variation of crank-pin effort is twofold:—Firstly, the variations are much more easily perceived and studied than in the polar diagram, and secondly, in the rectilinear diagram the area of the curve directly measures work done, being positive for loops above the base line MZ and negative for those below.

In the case taken the linear scale for the base line is 2 ins. = 1 foot, the crank-pin path amounting to just 5½ ft. per revolution; MZ thus represents a length of 11 ft., and is conveniently subdivided into 48 equal parts, each accordingly representing an angular movement of 15° of the crank-pin; angle is counted from the commencement of the working stroke.

For the tangential force scale, 0·04 in. = 1 lb. per sq. in. of piston area is taken; the area of the piston is 113 sq. ins.; hence 0·04 in. = 113 lbs. of tangential force at the crank-pin is the vertical scale of force in the rectilinear diagram.

The line of mean tangential force at the crank-pin during the cycle is next to be drawn; if the engine be running steadily under a constant external load, this is also the line of external load resistance, and is a straight line parallel to the base line MZ. If the external load be not constant, this line will not be straight, but curved, and must be specially determined from the nature of the varying external load; we here assume the case of a constant load.

The value of the mean tangential force may be obtained from the rectilinear diagram (*a*) by the usual method of ordinates, with due regard to the positive and negative work areas; or (*b*) by planimeter;

or again (*c*) from the mean effective pressure as determined from the indicator diagram. If determined by method (*c*) the procedure is as follows: In the case taken in fig. 240 the mean effective pressure from the diagram was found to be 82·5 lbs. per sq. in.; hence the useful indicated work per cycle is $82{\cdot}5 \times 113 \times 1{\cdot}75 = 16{,}314$ foot lbs. But in one complete cycle the crank-pin passes over 11 ft.; the mean effort is accordingly $\frac{16{,}314}{11} = 1483$ lbs.; HKN is therefore drawn at the height corresponding to 1483 lbs. on the vertical scale of force. Evidently HKN so divides the curve of tangential force that the sum of the areas of the loops lying above it is equal to the sum of those lying below. The very great inequality of the tangential driving force on the crank-pin in the case of a single-cylinder, single-acting, 'Otto' cycle engine is clearly exhibited in the rectilinear diagram; during the working stroke energy represented by the area HLK in excess of the mean energy is supplied by the working gases; during the exhaust and induction strokes the crank-pin force is at first negative and later positive; while throughout the compression stroke it is negative. The motion of the engine would thus be extremely irregular but for the addition of a large and heavy flywheel, by the inertia of whose rim, as already pointed out, the range in the cyclical speed variation may be reduced to any desired degree. In considering the flywheel energy, it is frequently sufficient to have regard to the heavy rim alone as providing the reservoir of energy; the additional inertia of the arms and boss in this case furnishes a margin in favour of increased uniformity in running. When desired, the energy of the arms and boss may be taken into account, and estimated as a fraction of that of the rim; Prof. Unwin ('Mach. Design,' ii. 170), for example, adds one-third of the mass of the arms and boss to that of the rim in calculating the energy of the wheel.

Coefficient of Fluctuation of Energy.—Referring to the rectilinear diagram of fig. 240, it is clear that while the crank-pin moves over the angular interval between H and K, energy represented by the area HLK is communicated to the engine in excess of the constant resistance against which it is, by supposition, working. This energy increases the angular velocity of the rotating parts of the engine; it may practically be regarded as absorbed by the flywheel rim alone on account of its large mass and radius. From K to H^1 at the commencement of the next cycle, the flywheel has to maintain the motion. Obviously, then, the velocity, and therefore the energy, of the wheel is a minimum at H and a maximum at K in each cycle.

The Coefficient of Fluctuation of Energy, $\frac{1}{K}$, is defined to be the ratio of the difference of the flywheel energy at maximum and minimum speed to the energy exerted in one complete cycle by the engine; thus we have for this case:

$$\text{Coefficient of fluctuation of energy} = \frac{1}{K} = \frac{\text{Area HLK}}{\text{Rectangle MN}} \tag{6}$$

Thus $\frac{1}{K}$ depends upon the nature of the cycle on which the engine works, on the number and disposition of the cylinders, and on the nature of the external resistances which the engine is called upon to overcome; when these data are known, the value of $\frac{1}{K}$ is determinable in the manner explained above. For the assumed single-cylinder, single-acting, 'Otto' cycle engine working against a constant resistance, as in Fig. 240, planimeter measurements furnish the value:

$$\frac{1}{K} = 1{\cdot}15 \tag{6A}$$

Now let ω_1, ω_2, and ω_0, denote respectively the angular velocity of the flywheel at maximum, minimum, and mean speed. Further, let E_0 denote the energy of the wheel at mean speed, and E the nett positive indicated work of the engine in one complete cycle. Then I denoting moment of inertia of the fly-wheel we have:

$$E^0 = \tfrac{1}{2} I \omega_0^2. \tag{7}$$

Also

$$\text{Area HLK} = \tfrac{1}{2} I(\omega_1^2 - \omega_2^2)$$
$$\text{Rectangle MN} = E$$

hence by Eq. (6)

$$\frac{1}{K} = \frac{\tfrac{1}{2} I(\omega_1^2 - \omega_2^2)}{E} \tag{8}$$

But we have

$$\begin{aligned}\tfrac{1}{2} I(\omega_1^2 - \omega_2^2) &= \tfrac{1}{2} I(\omega_1 - \omega_2)(\omega_1 + \omega_2)\\ &= I\,\omega_0 \,.\, \frac{\omega_1 - \omega_2}{\omega_0} \,.\, \frac{\omega_1 + \omega_2}{2}\end{aligned} \tag{9}$$

Now $\frac{\omega_1 - \omega_2}{\omega_0}$ is the ratio of the difference between the maximum and minimum angular velocity of the wheel to the mean velocity, and this is termed the Coefficient of Fluctuation of Speed, and is denoted by $\frac{1}{m}$. The value of this coefficient depends upon the nature of the external work done by the engine, and varies from about $\frac{1}{20}$ in cases of pumping, to as little as $\frac{1}{180}$ where electric alternators are to be belt-driven in parallel. Thus $\frac{1}{m}$ has a known value in any specific case, and is accordingly a known quantity in the formula.

Again, $\frac{\omega_1 + \omega_2}{2}$ is the mean between the maximum and minimum angular velocity of the flywheel; but, by the very *raison d'être* of the flywheel, ω_1 and ω_2 are not very different in value, and hence this mean is also practically the same as the mean speed of the engine, viz. ω_0; substituting therefore in (9), we have:

$$\tfrac{1}{2}\text{I}(\omega_1^2 - \omega_2^2) = \text{I}\omega_0 \times \frac{1}{m} \times \omega_0 = \frac{\text{I}\omega_0^2}{m}$$

$$= \frac{2\,\text{E}_0}{m} \text{ by Eq. (7)}$$

Hence, from Eq. (8):

$$\text{E}_0 = \frac{m}{2\text{K}} \cdot \text{E} \qquad (10)$$

an important result, expressing the energy of the flywheel at mean speed in terms of the Coefficients of Speed and Energy, and the energy exerted by the engine in one complete cycle.

We are now in a position to determine the necessary dimensions of flywheel rim for given values of K, m, and E.

For if HP be the indicated horse-power of the engine at n revolutions per minute, we have:

$$\text{E} = 66{,}000\ \frac{\text{HP}}{n} \text{ ft.-lbs.} \qquad (11)$$

Also, if R denote the mean radius of the flywheel in feet, and if W denote the weight of the rim in lbs., then, to a sufficient degree of approximation:

$$\text{I} = \frac{\text{W}}{g} \cdot \text{R}^2 \qquad (12)$$

And as $\omega_0 = \frac{2\pi n}{60}$, we have:

$$\text{E}_0 = \tfrac{1}{2}\text{I}\omega_0^2 = \tfrac{1}{2}\frac{\text{WR}^2}{g} \cdot \frac{4\pi^2 n^2}{3600}$$

i.e.

$$\text{E}_0 = 0{\cdot}00017\ \text{WR}^2 n^2 \qquad (13)$$

Substitute for E and E_0 from (11) and (13) in Eq. (10), and we obtain, after reduction:

$$\text{W} = 1{\cdot}983 \cdot \frac{m}{\text{K}} \cdot \frac{\text{HP}}{\text{R}^2 n^3} \cdot 10^8 \text{ lbs.} \qquad (14)$$

For example, in the case of fig. 240, we have $\frac{1}{K} = 1\cdot15$; HP $= 49\cdot5$; $n = 200$. Take $\frac{1}{m} = \frac{1}{150}$, then by (14) we get

$$W = \frac{206,850}{R^2} \text{ lbs.} \qquad (14A)$$

Now flywheels are most commonly of cast iron, and up to about 10 ft. in diameter are generally cast in one piece. Cast iron is weak and uncertain in tension, and moreover there are usually stresses of incalculable value within the wheel due to variations in the rate of cooling of the casting. For these reasons a limiting rim velocity determined by long experience must not be exceeded with such wheels in order to avoid risk of the disastrous results of a burst flywheel. In present practice, with stationary internal combustion engines, the rim velocity at normal speed is from 80 to 100 ft. per second; in the case of large built-up wheels a rim velocity of about 40 ft. per second only is usual. For one-piece cast-iron wheels we may take the velocity of the free extremity of the mean radius R in Eq. (14) as 85 ft. per second; then the maximum practicable value of R is determined by the engine speed alone from the relation $2\pi R n = 60 \times 85$, whence:

$$R_{\text{max.}} = \frac{815}{n} \text{ ft.} \qquad (15)$$

Thus, in the engine of fig. 240, $n = 200$, and thus $R = 4\cdot075$ ft; the mean diameter of the flywheel may thus be, say, 8 ft. 2 ins.

Hence, from (14A) the corresponding weight of the rim is, roundly, 12,450 lbs., or about $5\frac{1}{2}$ tons.

Now cast iron weighs about 468 lbs. per cub. ft., hence the volume of the rim must be $\frac{12,450}{468} = 26\cdot6$ cub. ft. The circumference corresponding to the mean radius of $4\cdot075$ ft. being $25\cdot6$ ft., the cross-sectional area of the rim must be $\frac{26\cdot6}{25\cdot6} = 1\cdot04$ sq. ft.

Thus a rectangular section 1 ft. 6 ins. wide by 8 ins. deep will suffice; so that finally for the engine of fig. 240, with a coefficient of fluctuation of speed of $\frac{1}{150}$, a suitable flywheel would be 8 ft. 10 ins. in external diameter, with a rim of rectangular section 18 ins. wide and 8 ins. deep; this result is in accord with current practice, but it should be remembered that the rim alone has been considered, and hence the speed fluctuation will in reality be less than $\frac{1}{150}$; by taking the inertia of the arms and boss into account the dimensions of the wheel may be somewhat reduced; in this connection it may be remarked that for internal combustion engine practice, the arms and boss may usually be taken as providing 0·15 of the whole inertia of the wheel, so that the weight of the rim alone need not be more than about 85 per cent. of W, as estimated by the above formula, when the arms and boss are taken into account.

Reverting to Eq (14), if A denote the cross-sectional area of the rim in square feet, then for cast-iron wheels we have:

$$W = 468 \times 2\pi RA \qquad (16A)$$

Substituting this value of W in (14) we obtain after reduction:

$$A = 6{\cdot}59 \cdot \frac{1}{K} \cdot \frac{m\text{HP}}{(nR)^3} \cdot 10^4 \text{ sq. ft.} \qquad (16)$$

One further simplification may be noted: If the flywheel be of cast iron, in one piece, and of maximum diameter, we have from Eq. (15) $R = \frac{815}{n}$; substituting this value of R in (16), it appears that for such cases:

$$A = 0{\cdot}000122 \, \frac{m\text{HP}}{K} \text{ sq. ft.} \qquad (17)$$

In the foregoing investigation it has been assumed that a working impulse occurs once in every two revolutions, but if, for example, an engine be governed on the 'hit-or-miss' principle, one or more working explosions may be missed when running below full load.

If, in such case, it be still necessary that the speed be maintained uniform within certain assigned limits, it will be necessary to estimate the value of the coefficient of fluctuation of energy, $\frac{1}{K}$, from the deficiency areas of the rectilinear diagram of effort, the sum of these then being considerably greater than the excess energy loop HLK; they consequently become the dominating factor in the evaluation of the energy coefficient.

Having obtained the rectilinear diagram of crank-pin effort for a single-cylindered engine, it is an easy matter to construct corresponding diagrams for engines having several cylinders, and in figs. 241 and 242 the characteristic curves for engines having two, three, four, and six cylinders, all of the single-acting 'Otto' type, are exhibited. Two cases require consideration with two-cylindered engines, viz.:

(1) Where the cranks are 180° apart, and cylinders together.

(2) Where the cranks are 0° apart, and cylinders together.

Case (1) is shown in the upper view of fig. 241; here two working impulses occur in one revolution, the following revolution being

CRANK-EFFORT CURVES FOR TWO-CYLINDER SINGLE-ACTING OTTO ENGINES.

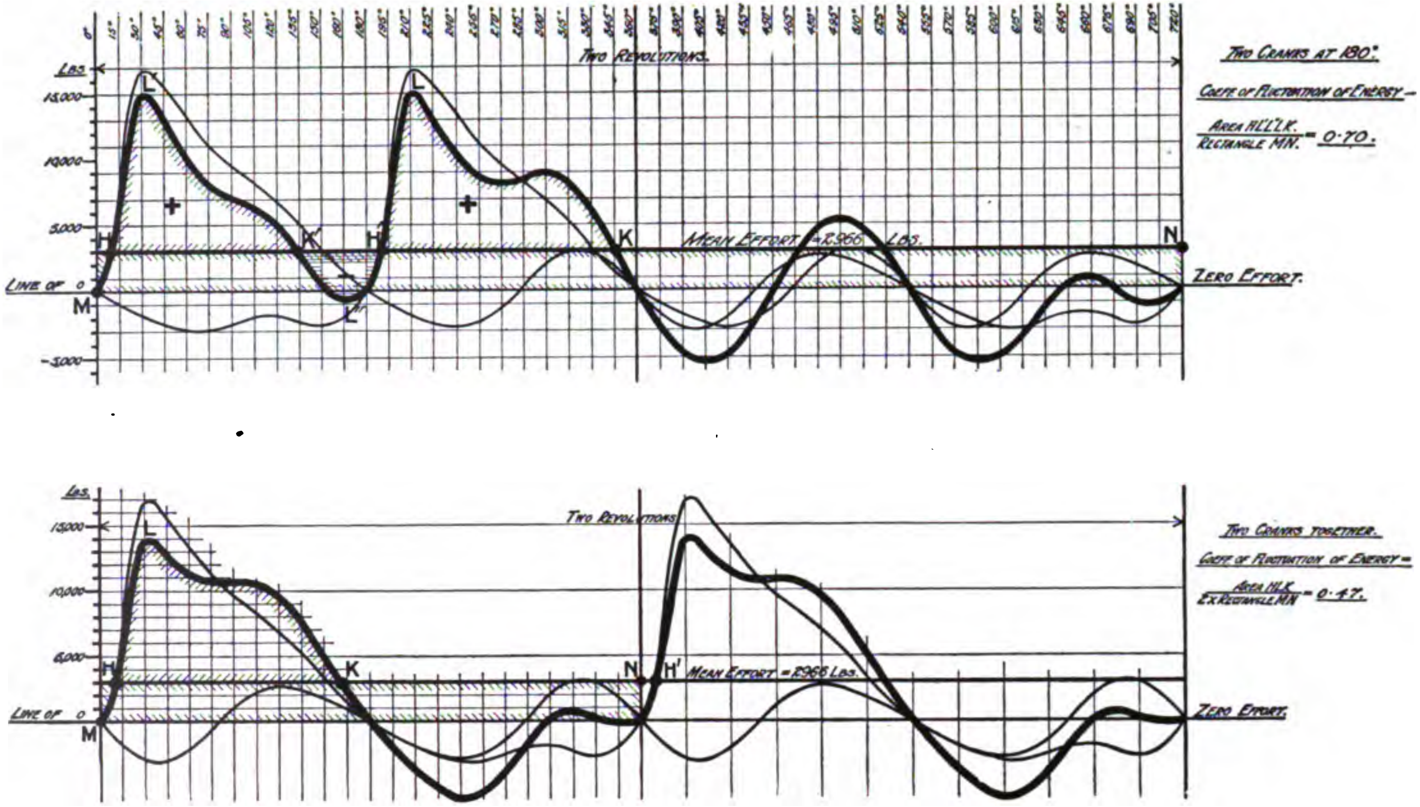


FIG. 241

idle; the turning effort is thus very un-uniform with this arrangement. The heavy-lined curve of resultant turning effort is obtained from two rectilinear curves, as on fig. 240, one of which is 180° in advance of the other; these are indicated in fine lines in fig. 241; the resultant curve is then found by taking the algebraical sum of the ordinates of these two curves at each point of construction. The vertical force scale is as on fig. 240; readings from the resultant curve, multiplied by the crank length in feet, give the crankshaft torque in lb.-feet.

The mean effort at the crank-pin during one cycle (two revolutions) is now twice as great as before; also, in determining the value of the coefficient of fluctuation of energy, $\frac{\text{I}}{\text{K}}$ it should be carefully noted that the 'numerator area' in this case must be HL′K′L″H′LK. For it is clear that the flywheel energy is a minimum at H; at K′ the increase in its energy is measured by the area KL′K′; at H′ this increase has been diminished by the small amount K′L″H′, but between H′ and K the large addition H′LK is made; evidently, then, the flywheel energy is at a maximum at K. The coefficient of fluctuation of energy is accordingly:

$$\frac{\text{I}}{\text{K}} = \frac{\text{Area HL}'\text{K}'\text{L}''\text{H}'\text{LK}}{\text{Rectangle MN}}$$

and in the particular case assumed in fig. 241 this ratio has the value 0·7.

Case (2) is exhibited in the lower figure of fig. 241; when the cranks are together (i.e. at 0°) there is a working impulse once in every revolution, and in consequence the inequality in the turning effort is somewhat lessened; the two curves as on fig. 240 are now placed 360° apart, and the algebraical sum of their ordinates furnishes the heavy-lined resultant curve shown; it will be noted that the amplitude of the resultant curve is here 360°, and not 720° as in the two preceding cases.

The flywheel energy is a minimum at H and a maximum at K; the coefficient of fluctuation of energy is accordingly:

$$\frac{\text{I}}{\text{K}} = \frac{\text{Area HLK}}{2 \times \text{Rectangle MN}}$$

Twice the rectangle MN is taken as the denominator, since in the preceding investigation E is, throughout, the energy developed by the engine in *two* consecutive revolutions; if we take the rectangle MN only as the denominator, $\frac{\text{I}}{\text{K}}$ will appear at twice the value, while E will correspondingly be reduced by one-half; this will make no change in the results if consistently employed, since the product $\frac{E}{K}$ is unchanged in value, and thus, Eq. (10), E_0 remains as before.

As estimated above, $\frac{I}{K}$ has here the value 0·47.

Fig. 242 exhibits turning effort curves for cases of engines having three, four, and six cylinders respectively.

The uppermost figure shows the curve for a three-cylinder engine with cranks at 120° ; the amplitude of the resultant curve will be seen to be 240°. The resultant curve is now the sum of three curves as on fig. 240 placed at intervals of 240°, as indicated in fine lines in the diagram. The coefficient of fluctuation of energy, $\frac{I}{K}$, is

$$\frac{I}{K} = \frac{\text{Area HLK}}{3 \times \text{Rectangle MN}}$$

and has the value 0·265 for the particular case assumed.

The central figure of fig. 242 exhibits the curve for a four-cylinder engine with the usual crank arrangement as indicated ; the construction of the resultant curve—which has now an amplitude of 180°—will be clear from the preceding remarks.

The turning effort is now much more uniform, and for the first time does not anywhere become negative ; it falls to zero at 180°, 360°, 540°, and 720°. As in Case (1) at fig. 241, it may be concluded that the flywheel energy is a minimum at H and a maximum at K, and accordingly that the coefficient of fluctuation of energy is :

$$\frac{I}{K} = \frac{\text{Area HLLLK}}{4 \times \text{Rectangle MN}}$$

and this, in the case taken, has the value 0·048 only.

The lowermost figure of fig. 242 shows the case for a six-cylindered engine with the usual crank disposition as indicated by the sketch on the right of the diagram ; the amplitude of the resultant dark-lined curve is now 120° only, and the turning effort is seen now to be always positive. The coefficient of fluctuation of energy is :

$$\frac{I}{K} = \frac{\text{Area HLK}}{6 \times \text{Rectangle MN}}$$

and, in the case taken, has the small value 0·03.

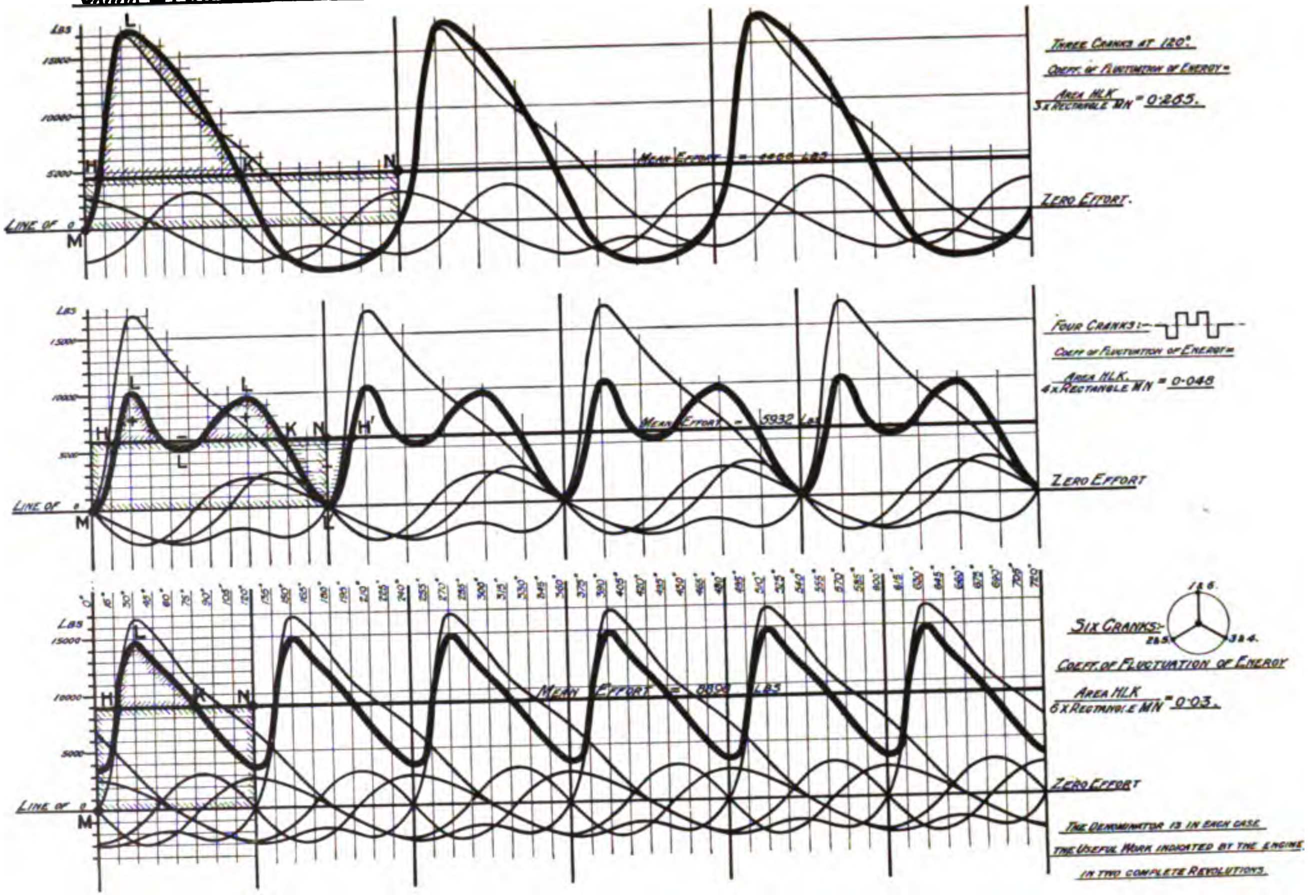
By means of the simple graphical processes above described and illustrated, the coefficient of fluctuation of energy may be determined for any given engine, regard being paid to any special circumstances of the case ; the results obtained above, together with some deductions therefrom, may be conveniently collected together in tabular form, and may prove of assistance as some guide in similar cases :

SOME DATA CONNECTED WITH FLYWHEEL DIMENSIONS SUITABLE FOR 12″ × 21″ SINGLE-ACTING OTTO CYCLE ENGINES RUNNING AT 200 R.P.M. RECIPROCATING MASS ASSUMED AS 4 LBS. PER SQ. IN. OF AREA OF PISTON. COEFFICIENT OF FLUCTUATION OF SPEED, $\frac{1}{m}$, TAKEN AS $\frac{1}{150}$ THROUGHOUT, AS IN THE DRIVING OF ELECTRICAL GENERATORS.

Number of cylinders	Ind. HP of engine	Fluctn. of energy		Mean crank-pin effort, lbs.	E in ft.-lbs. per two revolutions	E_0 in ft.-lbs. from Eq. (10)	Weight of			Cross-sectional area of rim in sq. in. (v. Eq. (17))	Wheel	
		Coeff. $\frac{1}{K}$ =	∴ K =				Rim, in lbs.	Rim, in tons	Whole wheel (app.), in tons		Mean radius in feet	App. external diameter, feet
1	49·5	1·15	0·87	1483	16,314	1,410,000	12,450	5·57	7¾	150	4·07	8′ 10″
2 at 180°	99	0·70	1·43	2966	32,628	1,710,000	15,100	6·75	9½	182	4·07	8′ 11″
2 at 0°	99	0·47	2·13	2966	32,628	1,150,000	10,200	4·56	6¼	122	4·07	8′ 9″
3	148·5	0·265	3·78	4450	48,942	972,000	8600	3·84	5¼	104	4·07	8′ 9″
4	198	0·048	20·8	5932	65,256	235,000	3840*	1·71	2⅜	63†	3·0	6′ 6″
6	297	0·030	33·4	8898	97,884	220,000	3600*	1·61	2¼	59†	3·0	6′ 6″

* For a mean flywheel radius of three feet; from Eq. (13).

† From Eq. (16A).



CRANK-EFFORT CURVES FOR THREE, FOUR AND SIX-CYLINDER SINGLE-ACTING OTTO ENGINES.

FIG. 242

In the first four cases of this table, the flywheel radius is at a maximum, and accordingly R and A are in these cases estimated by aid of Eqs. (15) and (17) respectively, and the rim weights have then a minimum value.

For the four and six-cylindered engines so large a value of R gives absurdly small values of the rim section, and accordingly in these two cases R has been assumed as 3 ft., and the rim area then estimated by aid of Eq. (16A).

It must be carefully noted that the flywheel dimensions given are for full load and *full speed* running, with *regular* and *equal* working impulses, and that the neglect of the inertia of the arms and boss furnishes a reserve in favour of increased uniformity of speed; the working impulses are, however, frequently very unequal, and the speed at full load not as high as that taken into consideration when designing; it will be noted from Eq. (14) that the weight of rim necessary for a given degree of cyclic irregularity varies inversely as the *cube* of the revolution speed, so that a comparatively small diminution of average engine speed may seriously reduce the speed regulating power of the flywheel.

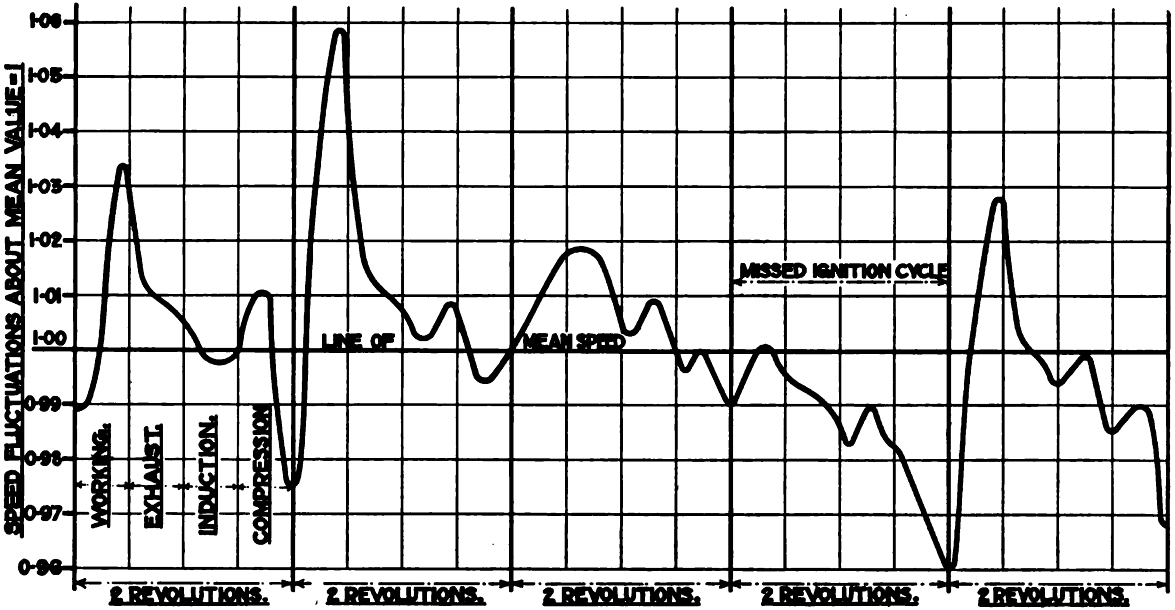
And again, as already remarked, if a definite degree of speed variation is to be maintained, although one or more consecutive explosions be missed, then a much more massive flywheel, or two wheels, must be used, the dimensions of which are ascertainable by the same general procedure as before.

According to Prof. Unwin, usual values of the coefficient of fluctuation of speed, $\frac{1}{m}$, are:

			$\frac{1}{m} =$
Engines	driving	pumps	$\frac{1}{20}$
”	”	machine tools	$\frac{1}{35}$
”	”	textile machines	$\frac{1}{40}$
”	”	spinning machinery	$\frac{1}{50}$ to $\frac{1}{100}$
”	”	electric-generators	$\frac{1}{150}$ to $\frac{1}{300}$

Actual tests of speed variation may be made by aid of a suitable recording tachometer as, for example, the ‘Moscrop,’ in which a small conical pendulum, very similar to the ordinary centrifugal governor, is so constructed as to move with a minimum of frictional resistance, and be very sensitive to changes in speed; the rising sleeve of this conical pendulum is connected to a lever, the end of which carries a stylus recording its position upon a paper band actuated by clockwork at an uniform rate in a direction at right angles to that of the motion of the stylus.

The conical pendulum is band-driven from the engine or machine whose speed fluctuations are to be examined, these being automatically



TACHOGRAM FROM SINGLE-CYL. SINGLE-ACTING OTTO CYCLE ENGINE. AT 60% LOAD.

FIG. 243

recorded upon the moving paper band by the stylus actuated from the sleeve of the sensitive conical pendulum ; in general, the rate of motion of the paper band is so slow that the stylus record appears as a dark band of varying breadth, which indicates the extremes of speed variation only.

By sufficiently increasing the velocity of the paper band, however, the time base on which the record is drawn is much extended, and in this way the fluctuations of speed occurring within individual strokes of the engine may be rendered manifest.

In fig. 243 an enlarged reproduction[1] is given from a tachometer record taken with a very sensitive instrument from a single-cylinder, single-acting, four-cycle Crossley engine of 100 horse-power, driven by producer gas. The normal full speed was 180 r.p.m., and the engine had two flywheels, each weighing 5 tons. At the time of the test, however, the horse-power was about 60 only, and the speed 164 r.p.m. ; at this reduced load the tachogram shows that the governor was causing the working impulses to vary considerably in magnitude, and occasionally an impulse to be missed altogether ; in consequence the cyclical fluctuations of speed were variable, and the total range of speed fluctuation shown considerable, ranging from a minimum of 0·96 to a maximum of nearly 1·06, the corresponding value of the coefficient of fluctuation of speed being :

$$\frac{1}{m} = \frac{1{\cdot}06 - 0{\cdot}96}{1} = \frac{1}{10}$$

At full load and full speed, with regular and equal working impulses, the cyclical fluctuations of speed would have been much less, as may be inferred from Eq. (14) by writing it in the form

$$\frac{1}{m} = \text{C} \cdot \frac{\text{HP}}{\text{WR}^2 n^3} \cdot \frac{1}{k} \qquad (18)$$

The increase in n and diminution in $\frac{1}{k}$ when at full speed and power combine to diminish $\frac{1}{m}$. Comparison of fig. 243 with the rectilinear diagram of tangential force at crank-pin given in fig. 240 shows that the tachometer record substantially confirms the results there obtained ; the speed reaches a maximum near the end of the working stroke, corresponding to the point K of the rectilinear diagram, and thereafter undulates down to a minimum at or near the end of the compression stroke agreeably with the fluctuations of energy exhibited in the crank-pin effort curve.

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[1] Re-drawn from Guldner's treatise, p. 217.

The problem of driving electric generators at a sufficiently uniform speed by few-cylindered, single-acting, four-cycle internal combustion engines has proved one of difficulty, and in order to allow for variations in the intensity of the working impulses and—with hit-or-miss governing—for missed explosions, it is frequently necessary to fit much heavier flywheels than would otherwise be necessary. Care must be taken to so design the governing gear and mixing and ignition apparatus that the explosions shall vary as little as possible in intensity and that, at reduced loads, the number of *consecutive* missed explosions shall be as small as possible.

Approximate Estimate of Rim-weight, W.—We may write Eq. (14) in the form:

$$\frac{W}{BHP} = c \cdot \frac{m}{R^2 n^3} \cdot 10^6 \qquad (14B)$$

lbs. of flywheel rim per BHP.

Mr. R. E. Mathot gives values for c for the various cases arising in practice; some of these, suitably converted to the units here employed, are given below:

Large four-cycle gas engines:

	c =
1. For single-cylinder, single-acting engines	265
2. For two single-acting cylinders side by side; cranks at 180°	167
3. For two single-acting cylinders opposed; cranks at 180°	125
4. For two single-acting cylinders side by side; cranks at 0°	125
5. For two single-acting cylinders opposed; cranks at 0°	167
6. For two single-acting cylinders tandem; cranks at 0°	125
7. For four single-acting cylinders opposed two and two; cranks at 0°	42
8. For single-cylinder, double-acting engines	167
9. For two double-acting cylinders side by side; cranks at 0°	42
10. For two double-acting cylinders tandem, on same crank-pin	42

Mathot gives also for *m* in the above equation the following values:

	m =
1. Industrial engines in ordinary workshops; wood-working and pumping machinery, &c.	30 to 40
2. Belt-driven continuous-current dynamos running in series	80
3. Belt-driven continuous-current dynamos running in parallel	120
4. Belt-driven alternators running in parallel	180
5. Cotton-spinning machinery	200
6. Direct-coupled alternators in parallel	250

The total weight of the flywheel he considers may be taken as about 1·4 W in usual practice.

On p. 181 (Chap. I) the total flywheel weight per BHP is tabulated for a series of large horizontal four-cycle gas engines of the double-acting tandem type; the mean is about 53 lbs., corresponding from Eq. (14B) to an average value of *m* of roundly 80. Flywheel weights of the two-cycle Mather & Platt Koerting engines and also of the Beardmore-Oechelhauser engines are exhibited in the table on p. 257 (Chap. II). For the Koerting engines the total flywheel weight per BHP ranges from 56 lbs. in the 400 BHP engine to 72 lbs. in the 1000 BHP. In the Oechelhauser engines the figure remains at the constant value of 112 lbs. over the same range of rated BHP.

METHODS OF GOVERNING

The methods adopted for the governing of internal combustion engines may be conveniently classified thus:

1. *Hit-or-miss Governing.*—Here when the engine speed exceeds the normal, the working charge is completely cut out, so that no working stroke occurs until the speed again falls to, or slightly below, its normal value.

2. *Quality Governing.*—In this case the governing gear reduces the proportion of gas, or other fuel, to air, leaving the mass of the charge per working stroke unchanged; as the engine load is reduced the mixture supplied accordingly becomes progressively weaker, while the compression continues unimpaired.

3. *Quantity Governing.*—In this method the degree of richness of the working mixture is preserved unchanged at all loads; but the quantity of the working charge admitted is reduced with the engine load; the compression thus diminishes with the load.

4. *Combinations* of quality and hit-or-miss, and of quantity and hit-or-miss, governing.

It may be remarked here that the petrol engines of automobiles are, in general, not now fitted with any special governing apparatus, speed control under varying load being by hand operation of the throttle; the ignition is also frequently arranged to be capable of advancement or retardation by hand, sometimes independently, sometimes conjointly with the throttle movement; independent control of the ignition is preferable. In many cases the throttle control is arranged to act also on the carburettor so as to preserve desired proportions of working mixture at all loads and speeds; reference to Chap. IX may be made in this connection.

1. *Hit-or-miss Governing.*—This method was for long almost universally used with stationary internal combustion engines, and is still adhered to by many English makers as, e.g. by the Crossley, Campbell, Fielding, National, Robey, Tangye, &c., companies, especially in the smaller engine sizes. It possesses the important advantage of giving economical fuel consumption at light as well as

at full loads, inasmuch as the number of working impulses is proportional to the load, and the quantity and quality of each working charge admitted to the combustion chamber is always the same; there are the further practical advantages that it is simple in mechanical detail, not liable to get out of order, and inexpensive to manufacture.

The method suffers from the defects that when one or more consecutive working impulses are missed the cyclic fluctuations of speed become considerable, and can only be brought within practicable limits by using very heavy flywheels; moreover, the working impulse following a miss or series of misses is frequently variable in intensity, being occasionally weaker, though usually much more powerful than the normal, in this latter case causing undue stresses in the engine.

For a coefficient of speed fluctuation, $\frac{1}{m}$, of about $\frac{1}{40}$, and for engines of up to about 100 BHP the simple hit-or-miss method of governing is still largely employed. With large engines the irregularity in the period of the working impulses is more serious, and there are also practical objections to the opening of large valves by means of knife-edged pieces. With the hit-or-miss method, the working charge is prevented from entering the combustion chamber in one or other of the following four ways:

(*a*) By not opening the gas or oil valve, so that air only enters during the suction stroke, thus cooling the cylinder walls.

(*b*) With automatic inlet valves, by holding the exhaust valve open; the cylinder walls in this case are not cooled to the same extent as before.

(*c*) With automatic inlet valves, by not opening the exhaust valve; the cooling of the combustion chamber is thus largely prevented; this is a desideratum in many oil engines in order to prevent deposition of the oil vapour in the working charge upon cooled surfaces.

(*d*) By not opening the inlet valve, thus causing the creation of a partial vacuum within the cylinder during the suction stroke.

With (*a*), on account of the scavenging action of the fresh air, the mixture first formed in the cylinder after a series of missed working strokes is at lower temperature than usual, mainly because of the replacement of the residual exhaust gas in the combustion chamber by air, and is, moreover, not heated to the same extent by the hot enclosing metal walls, so that the mass of the charge exceeds the normal; the resulting explosion pressure is accordingly sometimes considerably greater than that attained under regular full load working conditions; method (*b*) is not so open to this objection.

2. *Quality Governing.*—Variation in the proportions of the working charge may be effected in the four following ways:

(*a*) By varying the duration of opening of the gas or oil valve during the suction stroke, the air admission remaining constant.

(*b*) By throttling the gas supply throughout the suction stroke, with constant air admission. In many recent designs the gas valve is not opened until part of the suction stroke has been made ; in this way a rich and readily ignitible mixture is ensured near the ignition points in the combustion chamber.

(*c*) With automatic inlets, by delaying the closing of the exhaust valve so that some exhaust gas returns into the cylinder during the first part of the suction stroke, diluting the working mixture entering during the latter portion.

(*d*) With automatic gas or oil valve, by varying the duration of opening of the air inlet valves.

In quality governing, as the mass of the working charge remains fairly constant, the compression is nearly the same at all loads, and hence the efficiency, so far as it depends on the compression, is not prejudiced at light loads. Against this advantage, however, are the practical drawbacks that the weak mixtures produced at light loads are difficult to ignite, and burn but slowly ; the heat losses are then considerably increased, and the combustion of the charge is often incomplete ; in some cases combustion is so slow as to continue to the end of the exhaust stroke and ignite the next incoming charge ; backfires produced in this way are more often observed with quality-governed engines than others. The more recent practice of delaying the opening of the gas valve until part of the suction stroke has been performed, thus giving a richer mixture than the average in the neighbourhood of the ignition points, largely overcomes the earlier difficulties in obtaining brisk ignition and sufficiently rapid combustion.

Quality governing is frequently adopted in large gas engines in cases where the load and speed are subject to comparatively little variation as, e.g. when driving the machinery of corn mills, spinning and weaving factories, and central station dynamos.

3. *Quantity Governing.*—Variation of the working charge by varying its mass, the proportion of gas to air in the entering charge remaining constant, is effected :

(*a*) By throttling the charge throughout the whole suction stroke.

(*b*) By varying the instant of closing the inlet valve.

(*c*) By forcing a part of the charge back into the suction pipes.

In quantity governing the reduction in the mass of the charge at light loads causes a reduction in the compression pressure, and hence some diminution of thermal efficiency at such times ; on the other hand, as the compression and expansion curves rise and fall together, the variation of crank-pin effort is not unfavourably affected. With poor gas, of small hydrogen content, high compression pressures

(175 lbs. per sq. in. and upwards) may be safely used, thus giving increased economy and more rapid and complete combustion of the charge, and it is especially in such cases that quantity governing is most satisfactory, since the compression at light loads then remains sufficiently high to ensure ignition of the then less compressed mixture.

The mechanical details of quantity governing are in general simpler than those necessitated by quality governing, and at light loads the frictional resistances in the engine are diminished due to the diminished compression; on the whole the balance of practical advantage in general favours governing by this method rather than that of quality.

4. *Combinations* of hit-or-miss with quality or with quantity governing.

If instead of the usual single V-grooved block of the hit-or-miss arrangement, a stepped block as shown at B (fig. 244) be provided having several V-grooves (usually about three), the pecker A will move

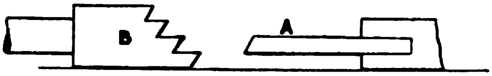

FIG. 244

the valve-spindle with which B is connected later and also through a smaller distance as A is raised by the governing mechanism; finally the pecker will wholly miss the block, and in this case the spindle remains stationary and a working stroke is cut out. If the spindle be that of the fuel valve, the combination is one of hit-or-miss and quality governing, while if it be the inlet valve spindle, the governing is by hit-or-miss and quantity.

Simple hit-or-miss governing has been very largely used up to the present time, particularly in Great Britain; Messrs. Crossley, for example, still (1911) fit it to all their engines up to about 100 BHP, though they have recently adopted the variable admission method as described later.

The arrangement adopted by the National Gas Engine Co., Ltd., is illustrated and described in Chap. I of this work; it ingeniously combines quality, quantity, and hit-or-miss governing in such manner as to obtain the leading advantages attaching to each of these three methods.

In gas engines, governing by variable admission (i.e. quantity governing) seems likely to replace the older methods in the near future; with oil engines the most convenient and satisfactory method in many cases is to govern by varying the charge of oil in inverse proportion to the piston speed, i.e. by quality governing.

Governing Arrangements

A type of governor much used in the past in conjunction with the hit-or-miss method is that known as the ' Inertia ' governor and, in a certain class of design, also as the ' Pendulum ' governor.

The ' pendulum ' governor was first used by Messrs. Crossley Bros. in 1885; it was the invention of the late Mr. H. P. Holt, and is fully described in his patent of that year. The device as applied to a small vertical Crossley engine is shown at fig. 90, p. 231 of the eighth edition of 'The Gas and Oil Engine.' Though now quite abandoned for all but the very smallest powers, it has been applied to some very large engines, M. Delamere-Deboutteville having even used it in the earlier 1000 horse-power Simplex type. A simple and ingenious arrangement —the invention of Mr. C. W. Pinkney—used in the smaller Tangye engines is diagrammatically shown in fig. 245. The pecker P is carried by a holder, A, hinged on the pin B and resting upon a roller turning on a fixed pin, C; B is borne in a bracket, D, to which a reciprocating motion is given by the eccentric-driven rod E E. The lower edge of A is formed with an inclined surface, K, as shown. H contains a spring by which an adjustable slight pressure is maintained upon the upper edge of A, thus tending to preserve contact with the roller C.

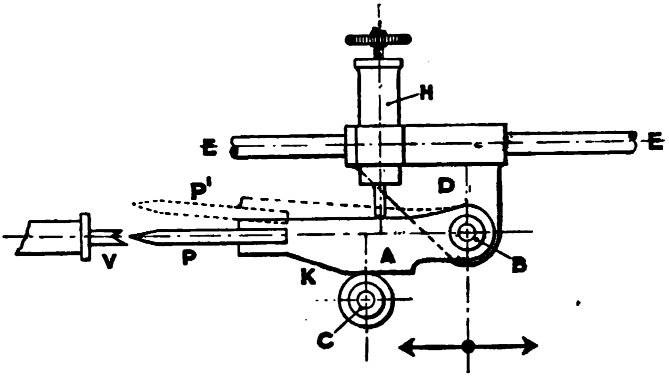

Fig. 245

At normal engine speed the spring pressure is so adjusted that A during its motion from right to left does not part contact with C, and the pecker P then engages with the groove V and the valve is opened. At increased engine speed, however, the upward momentum communicated to A by the reaction of the roller on the inclined surface K is sufficient to cause the pecker to take a position as indicated in dotted lines, and so long as this occurs the groove V is missed and the valve remains shut.

Another typical arrangement of the inertia governor or 'pendulum' governor is illustrated in fig. 246; this shows the disposition adopted in the 5 horse-power Crossley kerosene engine, running normally at 330 revolutions per minute. The pecker P here operates the exhaust valve spindle so that the exhaust does not open when the pecker misses the groove block; in this engine the vapour valve is automatic, and accordingly remains closed when the exhaust valve is not opened. The pecker P is carried by the bell-crank A A A capable of rocking upon the pin B borne in the upper end of the actuating lever L L; the upper arm of the bell-crank is expanded into a relatively massive piece as indicated. M is an adjustable spring in compression by means of which the bell-crank A A A is normally kept in the position shown in the figure. The actuating lever turns about the fulcrum E, and is driven by the cam H and roller K; the cam gives periodic impulses, F, to the pin B, as indicated by the arrow. The impulse F is equivalent to an equal impulse, F′, at the centre of the pendulum mass, together with an impulsive couple, F F″, tending to cause clockwise rotation of the bell-crank upon the pin B; at normal full speed the compression of the spring M is so adjusted that this turning tendency is just resisted, and the pecker then engages its notch at every impulse of the lever L; if, however, the engine speed increases, the couple F F″

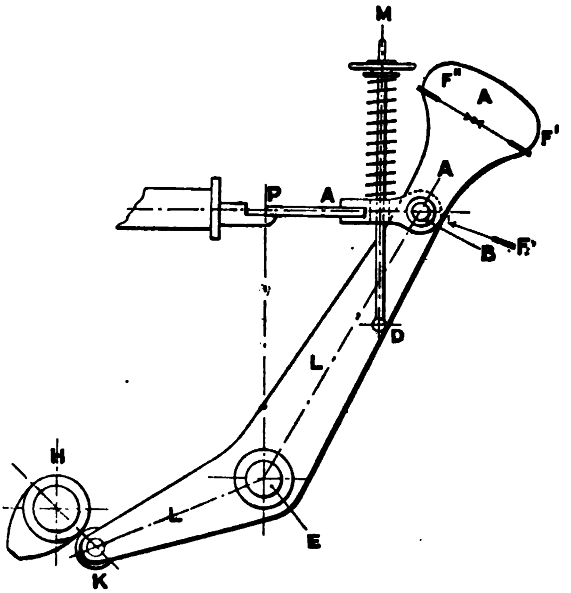

Fig. 246

overcomes the spring resistance and causes the bell-crank A A A to turn slightly about B in a clockwise sense, thus causing the pecker to miss its notch, and consequently cutting out the working stroke; this continues until the speed of the engine is reduced.

In fig. 247 is illustrated another inertia governing arrangement as used in the smaller sizes of the Stockport gas engine.

Here a cast-iron cylindrical weight, M, is capable of sliding freely on a vertical rod, C, carried in the bracket D, which is attached to the rocking lever L; the weight is supported on a spring of which the compression may be adjusted by means of the lock nuts as shown.

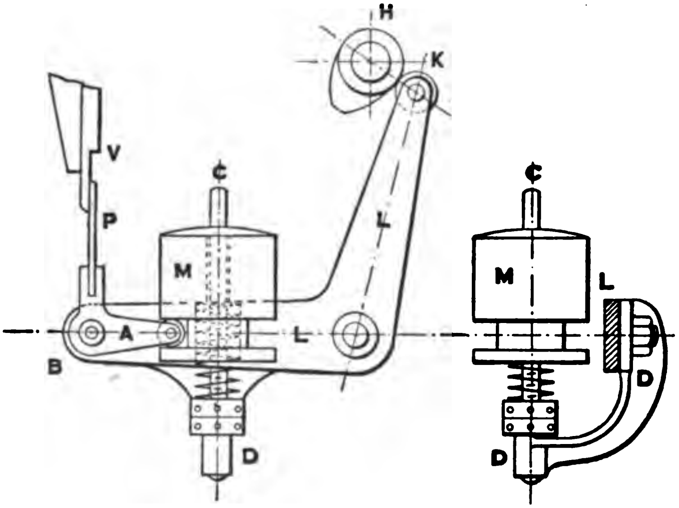

FIG. 247

The groove in M engages the roller end of the bell-crank A, which is borne on a pin, B, in the end of the rocking lever L, and carries the pecker P at its other extremity; L is actuated through the cam H and roller K. So long as the lever L and cast-iron weight M move together the pecker engages the notch V and the valve is opened; when, however, the engine speed increases, the inertia of M causes it to lag behind, thus depressing the roller end of the bell-crank A and causing the pecker to miss the notch; this continues until the speed is again reduced.

Occasionally in hit-or-miss governing the pecker has been operated electrically as, for example, in Kilmarnock's arrangement, wherein a small centrifugal governor completes the circuit of an electromagnet when the engine speed exceeds the normal; this electromagnet by

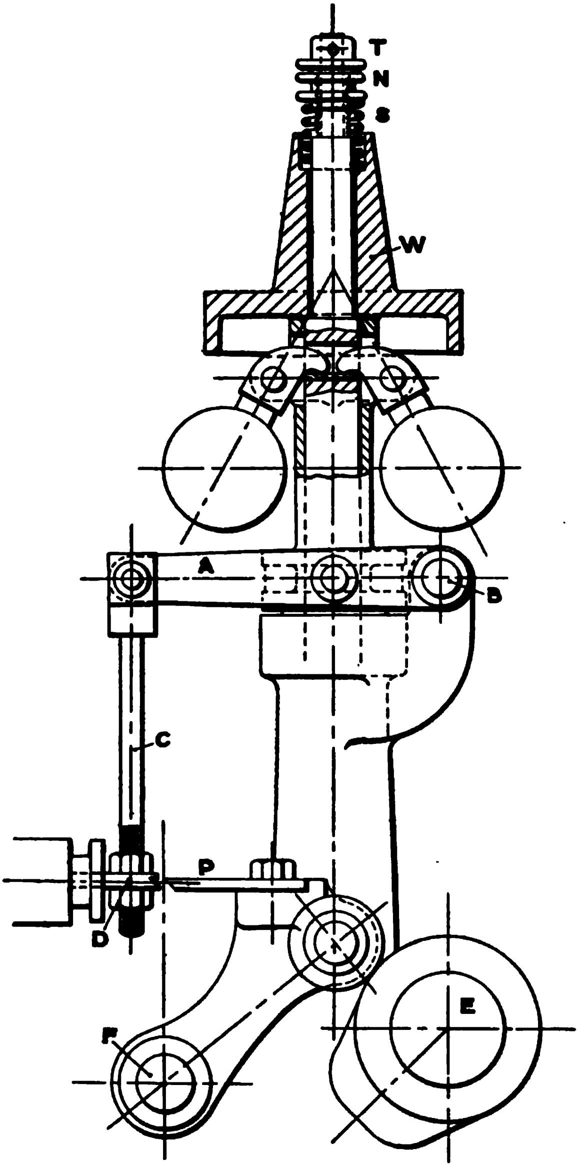

Fig. 248

attracting the pecker blade causes it to miss the block and thus cuts out the working stroke.

But in the very great majority of cases where hit-or-miss regulation is retained a centrifugal governor usually with vertical, though occasionally with horizontal, axis is employed to move the pecker block at speeds beyond the normal; a characteristic arangement is illustrated in fig. 248, which shows the disposition of parts as used in their smaller engines by the National Gas Engine Co., Ltd. The pecker P normally operates the gas valve through the pecker block D, suspended by a rod, C, from the free end of a lever, A, pivoted at B; a very small lift of the governor balls thus suffices to raise the block D out of the way of the pecker. The governor is driven at about 275 r.p.m. from the half-speed shaft E by helical gearing, and is loaded by a cast-iron cap, W, and spring S; the degree of compression of S can be varied by the neat arrangement of screw and nuts N while the engine is running, and the governor action thus regulated to a nicety; the collar T provides the abutment for S and N. Fig. 249 shows diagrammatically in plan the arrangement of pecker and block in the 60 horse-power Crossley gas engine. The pecker blade is permitted a slight amount of rock to right or left in its holder, and is supported by a block, E, with a shallow V-groove, as shown in the sectional elevation, thus ensuring its correct engagement with the pecker block B which, in this design, has two V-grooves as shown. This pecker block is drawn towards the right by the governor, through the rod C, when the engine speed increases, thus missing the stroke of the pecker. The two-ball spring-controlled governor is here placed with its axis horizontal, and being direct driven from the crankshaft a more uniform rotation is obtained than by the usual method of driving from the camshaft.

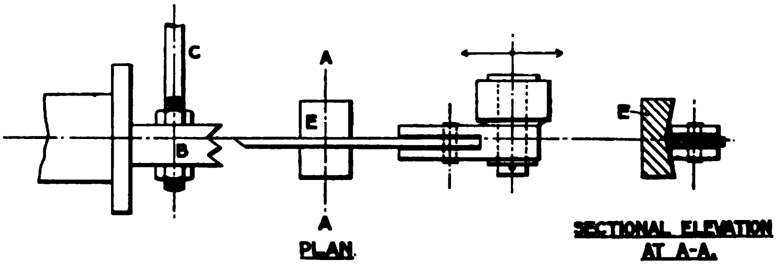

FIG. 249

In the crude oil Crossley engine (*v.* Chap. X) the vertical centrifugal governor is driven from the camshaft, and the engine is governed by varying the quantity of fuel oil delivered into the combustion chamber, with complete cut out at very light loads.

The arrangement adopted is illustrated in fig. 250; the bell-crank A is driven through the V-grooved block B, suspended from the governor rod C. The pecker has here three blades arranged as shown; these are carried in a bracket, E, pivoted about H, and driven through a roller, L, by the cam K. The fuel oil pump plunger is D; immediately behind this, and operated also by A, is the water-injection pump plunger, the stroke of which thus varies with that of the fuel oil pump. The speed of the engine determines, through the governor, the height of the block B, and thus the particular pecker blade which engages the V-groove; the stroke of the fuel oil pump, and therefore the quantity of oil injected into the combustion chamber, is accordingly diminished as the speed of the engine increases. If the speed be sufficiently increased, B is raised so high that the topmost pecker blade misses the groove; in this case no fuel oil is injected, and the engine misses a working stroke; it will be noted that no water spray then enters the cylinder either, the two plungers being simultaneously operated by A.

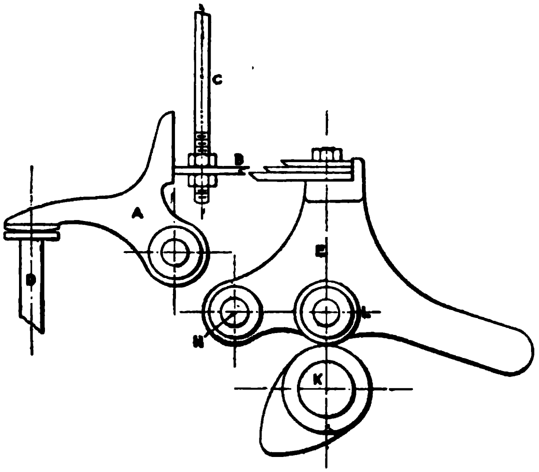

Fig. 250

In the Fielding & Platt kerosene engine (*v.* Chap. X, fig. 421) a similar arrangement obtains, but there is here only one pecker blade and a single V-grooved block; the governing is accordingly simple hit-or-miss, the engine receiving either a full charge of vapour or none. The bell-crank A in this design operates both the kerosene and water-injection pumps and also the vapour valve; hence when a miss occurs the vapour valve remains closed and air alone is drawn into the cylinder through the separate air valve shown in the figure referred to.

The Campbell Co. employ hit-or-miss governing in all their oil engines, the governor acting on the exhaust valve. A centrifugal governor, weight-loaded, is used with a very light regulating spring attached to the sleeve lever for adjusting the speed to the desired rate. These oil engines are built in single-cylinder horizontal units up to 70 BHP and in double-cylinder units to 140 BHP. Hit-or-miss governing is also used in the Campbell gas engines up to about 25 BHP; for higher powers 'quality' governing is adopted in the horizontal types and 'quantity' in the multi-cylindered vertical types; in these larger gas engines the 'Hartung' design of governor (*v. infra*) is employed.

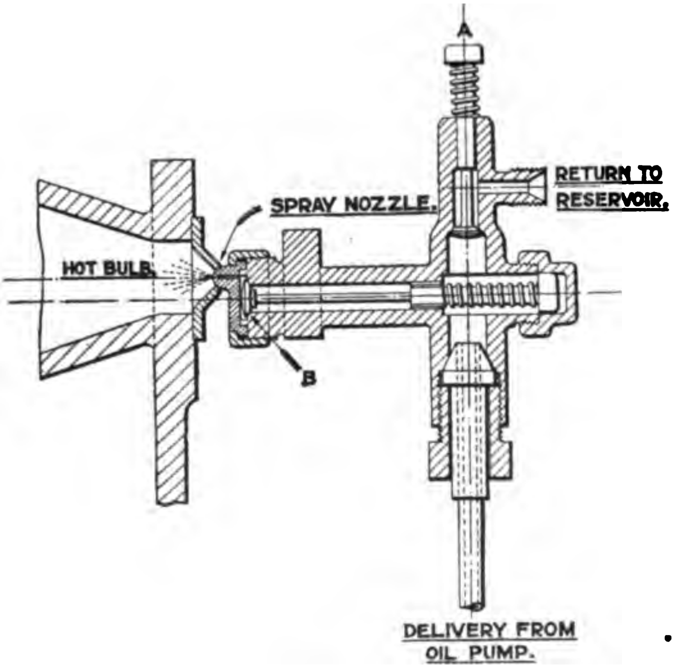

FIG. 251

As already stated, Messrs. Tangye use the hit-or-miss method with inertia governor for powers up to about 60 BHP; recently, however, they have adopted the quantity method for engines above about 15 BHP; here again the 'Hartung' type of governor is employed. In the Hornsby oil engines a loaded or 'Porter' governor acts on a small by-pass valve in the fuel oil delivery pipe, thus permitting part of the charge pumped to return to the oil reservoir. The arrangement is shown in fig. 251; at full load all the oil pumped passes the valve B and enters the hot bulb through the fine spraying orifice indicated;

when the load is reduced and the engine speed increases the governor depresses the spindle of the valve A, and part of the oil pumped then returns to the reservoir by the branch shown.

In the smaller Hornsby-Stockport gas engines the governing is by hit-or-miss acting on the gas valve, the spring-controlled centrifugal governor having its axis horizontal; in the larger sizes the governing is by a combined quality and quantity device, such that at light loads the compression is still about 75 per cent. of that at full load, in order to obtain reasonable economy and greater certainty of regular ignition when running light.

Governors

With hit-or-miss regulation the design of the governor is, in general, simple, very little power being required, and the sleeve travel to cause the necessary motion of the pecker block being also small; thus in the case illustrated in fig. 248 the total travel of the sleeve is only about one-eighth of an inch. With quality or quantity governing, however, the governor must have a greater range and more power, and accordingly it may be useful to consider briefly here some of the essential characteristics of the centrifugal governor or 'Conical pendulum.'

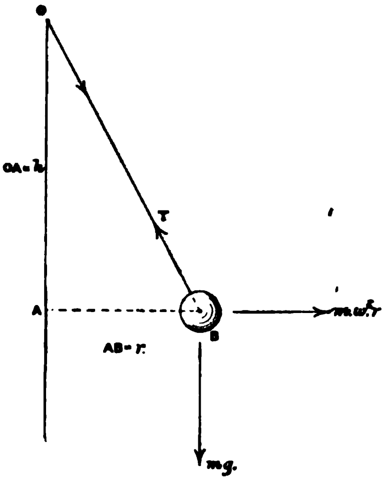

FIG. 252

The simple conical pendulum consists of a mass, B, suspended by a light inextensible string from a point, O, in a vertical axis about which B revolves in a horizontal circle of radius AB. Let m be the mass of B in lbs.; let OA $= h$; AB $= r$; both in feet; and let the angular velocity of B be ω radians per second. Then B is in equilibrium under the action of its weight, mg poundals, acting vertically downwards; the centrifugal force $m\omega^2r$ poundals acting horizontally outwards; and the tension, T poundals, in the string. To these three forces the sides of the triangle OAB are respectively parallel, and hence, by the triangle of forces, we have:

$$\frac{mg}{m\omega^2r} = \frac{\text{OA}}{\text{AB}} = \frac{h}{r},$$

and consequently

$$h = \frac{g}{\omega^2} \text{ ft.} \tag{19}$$

Thus the height of the conical pendulum is independent both of the mass of B and of the length, OB, of the string, and depends only upon the angular velocity of the bob about the vertical axis.

Observe that if τ be the time of one revolution in seconds, then as $\tau = \frac{2\pi}{\omega}$, we have by (19)

$$\tau = 2\pi\sqrt{\frac{h}{g}} \tag{20}$$

which is the time of vibration of a simple oscillating pendulum of length h ft. If n be the number of revolutions per minute corresponding to an angular velocity of ω radians per second, then, as $n = \frac{60\omega}{2\pi}$, we may write (19) in the form :

$$h = \frac{2936{\cdot}3}{n^2} \text{ ft.} \tag{21}$$

This is sometimes a useful quantity in calculations relating to governors, and is styled the 'Height due to the revolutions'; its value in feet and in inches, for several values of n, is exhibited in the following short table :

HEIGHT DUE TO REVOLUTIONS AND HEIGHT VARIATIONS, FROM EQS. (21) AND (22A)

Revs. per minute, n	h in		Δh in inches from Eq. (22A)
	Feet	Inches	
0	∞	∞	—
50	1·1745	14·094	2·11
100	0·2936	3·523	0·528
150	0·1305	1·566	0·235
200	0·0734	0·881	0·132
250	0·04698	0·564	0·084
300	0·03262	0·391	0·059
350	0·02397	0·287	0·043
400	0·01835	0·220	0·033
∞	0	0	—

It will be observed that h rapidly diminishes as the revolution speed increases ; differentiating Eq. (21) with respect to n, we have :

$$\Delta h = -\frac{5872{\cdot}6}{n^3} \cdot \Delta n \qquad (q/p) \tag{22}$$

The minus sign indicates that h diminishes as n increases.

In practice it is frequently necessary that if the engine be suddenly relieved of full load the momentary increase of speed shall not be more than 7½ per cent. of the normal, with a rapid reduction to a steady light-load speed of 2½ per cent. only in excess of the normal; suppose then, in Eq. (22) we take Δn to be 7½ per cent. of n, that is $\frac{\Delta n}{n} = 0{\cdot}05$; the corresponding diminution in h is then

$$\Delta h = -\frac{440{\cdot}4}{n^2}\text{ ft.} \tag{22A}$$

and values of Δh (in inches) calculated from this result are given in the right-hand column of the table above; it will be seen from these figures that the diminution in h for a 7½ per cent. increase in speed is a very small quantity for revolution rates of 200 per minute and over.

Now the two-ball governor derives its ability to operate the speed regulating gear of the engine from the centrifugal force of the balls, and this force varies directly as their mass, directly as the radius of the circle in which they revolve, and directly as the square of their revolution rate.

Practical considerations require that the governor shall not be a very bulky and heavy adjunct, and hence m and r are kept small; in order that the centrifugal force may still be adequate it becomes necessary to increase ω, i.e. to 'speed up' the governor. This is effected, without diminishing h, by loading the governor sleeve either by a dead weight or by spring compression, or by a combination of these two methods.

Most usually in governors the points of suspension of the arms carrying the balls are not in the axis of revolution, as shown at A in fig. 253, but at a short distance therefrom, as at D D′ in B; the apex of the cone of revolution is then the point O, the length OA being h; when the balls rise, the height of the cone diminishes both from its base and apex; thus in case B we have $\Delta h = \text{OA} - \text{O}'\text{A}'$.

A small change of speed causes a large movement of the balls in the Farcot crossed-arm governor illustrated in fig. 253, case C; as the balls rise the cone of revolution diminishes in height from the base, but increases at the apex, and in this case we have $h = \text{OA}$ and $\Delta h = \text{OA} - \text{O}'\text{A}'$. The three cases of fig. 253 are drawn for the same h and Δh, and hence for the same change from the same speed; it will be seen that the lift in case C is markedly greater than in cases A and B.

If, in case C, the point of suspension be taken farther from the axis of revolution, a position is soon reached where Δh is sensibly zero over the range of lift of the balls; these will then pass from their lowest to their highest position for an extremely small change in the

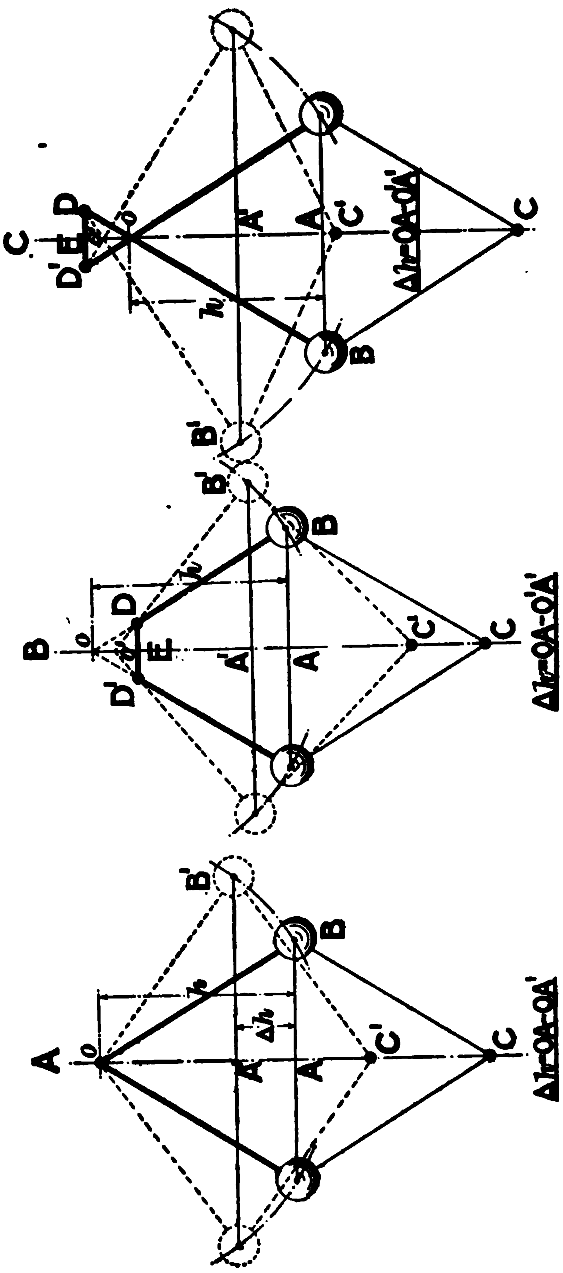
A
o
h
Δh
A'
A
C'
C
B
B'
Δh=OA-OA'
B
o
D
D'
E
h
A'
A
C'
C
B'
B
Δh=OA-O'A'
C
D'
E
D
o
h
A'
A
C'
C
B
B'
Δh=OA-O'A'

FIG. 253

revolution speed. By still further removing the points D and D′ from the axis, h may be caused to *increase* as the balls rise, the governor in this case becoming unstable.

In Eq. (19) suppose h of constant value ; then ω also is constant.

That is, if a conical pendulum be so contrived that its height be constant, then it has only one revolution rate.

Now (fig. 254) let the ball B move on a smooth parabolic guide whose axis is YY′ ; let B be at rest upon the curve when the whole is in rotation about YY′ ; then B is in equilibrium under the action of its weight, its centrifugal force, and the reaction of the curve, which, being

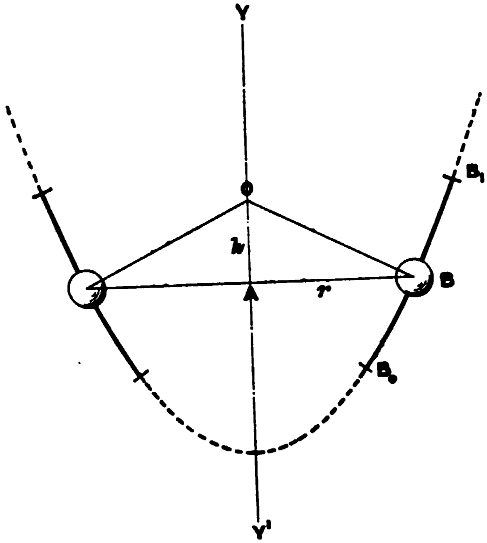

FIG. 254

supposed smooth, is in the direction of the normal BO. Thus BOA is a triangle of forces for the mass B, and accordingly, as before, we have $\frac{m\omega^2 r}{mg} = \frac{r}{h}$, whence $\omega^2 = \frac{g}{h}$.

Now in this case h = OA is the subnormal of the parabola, which is well known to be of constant length in this curve ; consequently ω has but one value if B is to remain on the curve ; any change in this value, however small, will in the absence of friction cause the balls either to fall to the bottom, B_0, or rise to the top, B_1, of their range of lift. An approximation to the parabolic path may be practically realised by taking the points D, D′, in case C of fig. 253 at such a distance from the axis that the circular arc in which B moves sensibly coincides with the part B_0 B B_1 of a parabola; to prevent risk of the governor becoming unstable the points D, D′ are actually taken rather nearer the axis than this, and will still give a very sensitive arrangement. Governors in which the balls remain in equilibrium only at one speed, the least change causing them to move to one or other limit of their range, are said to be isochronous; thus the parabolic gravity-controlled governor is, in the absence of friction, perfectly isochronous.

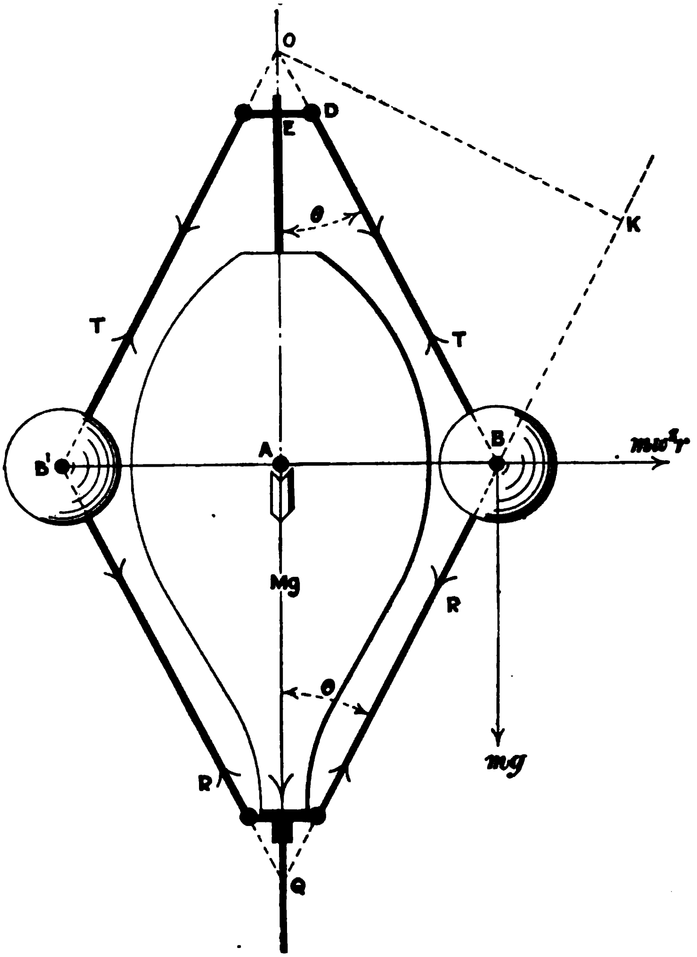

FIG: 255

Equilibrium of the Loaded Governor.—The loaded or 'Porter' governor is illustrated in fig. 255; the sleeve is here loaded with an axial dead weight of mass M lbs. as shown, and the usual case in which the four links are of equal length will be considered. The height of the cone is OA = H ft., while AB = r is the radius of the circle in which the balls move.

The ball B is in equilibrium under the action of:

(1) The centrifugal force, $m\omega^2 r$ poundals.
(2) Its own weight, mg poundals.
(3) The tension T in the link BO.
(4) The tension R in the link BQ.

Taking moments about O, we have:

$$m\omega^2 r\mathrm{H} = mgr + \mathrm{R} \times \overline{\mathrm{OK}} \tag{23}$$

Now the point Q is in equilibrium under the action of R along QB and QB′, and the weight Mg acting vertically; hence we have:

$$\frac{\mathrm{R}}{\mathrm{M}g} = \frac{\sin(\pi - \theta)}{\sin 2\theta} = \frac{\sin\theta}{2\sin\theta\,.\,\cos\theta} = \frac{1}{2\cos\theta}$$

so that

$$\mathrm{R} = \frac{\mathrm{M}g}{2\cos\theta}$$

Again,

$$\mathrm{OK} = \mathrm{OQ}\sin\theta = 2\mathrm{H}\sin\theta$$

hence

$$\mathrm{R} \times \overline{\mathrm{OK}} = \mathrm{M}g\mathrm{H}\tan\theta$$

but

$$\tan\theta = \frac{r}{\mathrm{H}}$$

thus, finally:

$$\mathrm{R} \times \overline{\mathrm{OK}} = \mathrm{M}gr$$

Substituting in Eq. (23), we have on reduction:

$$\omega^2 = \frac{g}{\mathrm{H}} \times \left(\frac{m + \mathrm{M}}{m}\right) \tag{24}$$

Now h being the 'height due to the revolutions,' we have by Eq. (19), $\omega^2 = \frac{g}{h}$; hence from (24) we have:

$$\frac{\mathrm{H}}{h} = \frac{m + \mathrm{M}}{m} \tag{25}$$

That is, in the loaded governor the height is greater than that due to the revolutions in the proportion of $(m + \mathrm{M})$ to m, where M is the mass of the axial dead weight, and m that of one of the revolving balls.

Again, if Ω be the 'revolutions due to the height H,' then, as $H = \frac{g}{\Omega^2}$ we have from (24):

$$\frac{\omega}{\Omega} = \sqrt{\frac{m + M}{m}} \qquad (26)$$

showing that in the loaded governor the actual revolution rate is greater than that due to the height in the proportion of $\sqrt{m + M}$ to $\sqrt{m}$.

Moreover, as $\frac{H}{h}$ is constant (Eq. 25), it follows that

$$\frac{\Delta H}{\Delta h} = \frac{m + M}{m} \qquad (27)$$

and thus the change in the height of the loaded governor for a given increment in the common speed is greater than that in the corresponding simple conical pendulum in the proportion of $(m + M)$ to m.

For example, if $\frac{M}{m} = 10$, and the speed be 300 r.p.m., we have, from the table on p. 374, $h = 0{\cdot}391$ in., and $\Delta h = 0{\cdot}059$ in., for a $7\frac{1}{2}$ per cent. increase. Hence, from Eq. (25):

$$H = h\left(1 + \frac{M}{m}\right) = 0{\cdot}391 \times 11$$
$$= 4{\cdot}3 \text{ ins.}$$

while from Eq. (27):

$$\Delta H = \Delta h\left(1 + \frac{M}{m}\right)$$
$$= 0{\cdot}65 \text{ in.}$$

results which exhibit clearly the practical advantages of loading.

The loaded governor is also often of the Farcot or crossed-arm type.

Instead of a dead weight the sleeve is often loaded by a spring; the sleeve load is then not constant, but increases as H diminishes, thus rendering the governor more stable but less sensitive. This method of loading has the advantage that the compression of the spring is easily adjustable. If the uncompressed length of the spring be L ft., and its resistance when compressed to a length l be P lbs. weight, then we may write:

$$P = k\left(1 - \frac{l}{L}\right) \qquad (28)$$

k being a constant and expressing, in fact, the load in lbs. weight that would compress the spring to zero length.

In the simple arrangement indicated in fig. 256 we have $l = 2H$, and hence for Mg in the preceding results we must substitute the expression $Kg\left(1 - \frac{2H}{L}\right)$; accordingly Eq. (24) becomes:

$$\omega^2 = \frac{g}{H}\left\{1 + \frac{k}{m}\left(1 - \frac{2H}{L}\right)\right\} \tag{29}$$

while Eq. (25) appears as

$$\frac{H}{h} = 1 + \frac{k}{m}\left(1 - \frac{2H}{L}\right) \tag{30}$$

Controlling Force.—The actual path of the ball being in a circle of radius r ft., with angular velocity ω radians per second, the resultant of all the forces on the ball must be a centripetal force equal to $m\omega^2 r$ poundals. This is termed the 'Controlling Force,' and denoting it by F, we have always

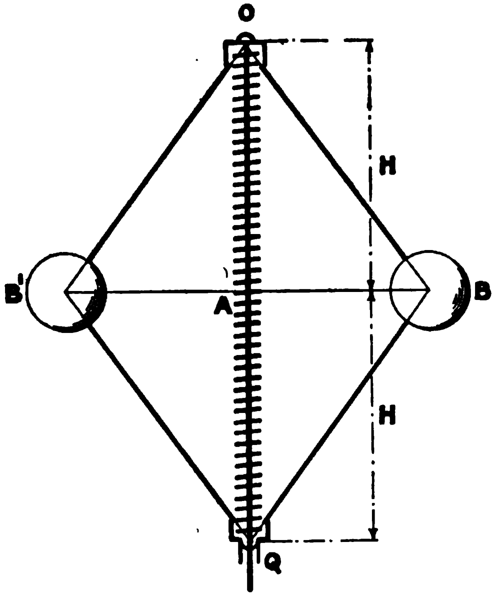

FIG. 256

$$F = m\omega^2 r = \frac{4\pi^2}{60^2} \cdot n^2 mr \tag{31}$$

whence

$$n = \frac{30}{\pi} \cdot \sqrt{\frac{F}{mr}} \text{ revs. per min.} \tag{32}$$

an equation giving the revolution rate in terms of mr and the Controlling Force.

Now F is always a function of the geometrical and dynamical arrangement of the governor, and is in general easily expressed in terms of r and constants of the apparatus.

For example, in the simple conical pendulum (fig. 252), if the length of the arm OB be a feet, we have $h = \sqrt{a^2 - r^2}$.

Hence

$$F = m\omega^2 r = \frac{mgr}{h} = \frac{mgr}{\sqrt{a^2 - r^2}} \tag{33}$$

and thus F is expressed in terms of r and the constants m, g, and a.

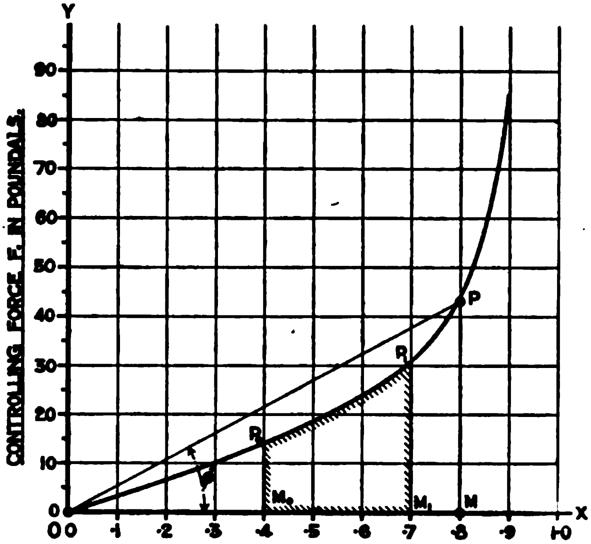

FIG. 257

Again, in the loaded governor, fig. 255, putting DB $= a$ and DE $= b$, we have:

$$H = AO = AE + EO = \sqrt{a^2 - (r - b)^2} + \frac{bH}{r};$$

i.e.

$$H = \sqrt{a^2 - (r - b)^2} \div \left(1 - \frac{b}{r}\right).$$

But

$$F = m\omega^2 r = \left(1 + \frac{M}{m}\right)\frac{mgr}{H} \quad \text{(Eq. 24)}$$

hence for this case

$$F = \frac{mg\left(1 + \frac{M}{m}\right)}{\sqrt{\left(\frac{a}{r-b}\right)^2 - 1}} \tag{34}$$

and is thus known in terms of r and constants; similarly other cases may be dealt with. Having expressed F in this way, it is instructive to plot a curve having values of r as abscissæ and corresponding values of F as ordinates; such curves are termed Curves of Controlling Force, and were first suggested by Mr. Wilson Hartnell in 1882. Thus, in fig. 257, the curve connecting F and r is drawn from Eq. (33), for $m = 1$ lb. and $a = 1$ ft. Draw a line OP from the origin to any point P on the curve, and denote the angle MOP by ϕ; then

$$\tan\phi = \frac{PM}{MO} = \frac{F}{r} = \frac{m\omega^2 r}{r}$$

i.e

$$\tan\phi = m\omega^2 \tag{35}$$

Thus the values of F and r corresponding to an assigned speed, ω, are at once obtained from the curve by drawing ϕ such that $\tan\phi = m\omega^2$ (having regard, of course, to the scales to which F and r are plotted), the corresponding controlling force F being given by PM, and radius r by OM.

Again, if the actual range in r be from say OM_0 to OM_1 (fig. 257), the range in the controlling force is from M_0P_0 to M_1P_1, and in speed from $\omega_0 = \sqrt{\frac{\tan\phi_0}{m}}$ to $\omega_1 = \sqrt{\frac{\tan\phi_1}{m}}$, where $\phi_0 =$ the angle M_0OP_0, and ϕ_1 the angle M_1OP_1.

Further, the whole work done in changing the configuration of the governor from r_0 to r_1 is evidently represented by the hatched area $M_0P_0P_1M_1$, and this is a measure of the 'powerfulness' of the governor; in symbols this is, of course, expressed by

$$\text{Work done} = \int_{r_0}^{r_1} F\,dr \tag{36}$$

In order that a governor may be stable, F must increase at least as rapidly as r over the range of action, as otherwise at some value of r the centrifugal force will become greater than F, and the balls will then fly out to the limit of their range.

A special case is when the controlling force F increases just in proportion to r; we then have $F = kr$, where k is some constant; but $F = m\omega^2 r$; hence, equating, we get for this case, $\omega = \sqrt{\frac{k}{m}}$, i.e.

ω is constant. Such a governor if run at any other speed than $\sqrt{\frac{k}{m}}$ will have the balls at one or other limit of their range, and is consequently, excepting for friction, perfectly isochronous. The curve of controlling force for such an isochronous governor is obviously a straight line, as OP passing through the origin (fig. 257); and hence a governor is unstable if, within its range of action, the curve of controlling force is at any point less steep than the line drawn from that point to the origin.

The stability of an isochronous governor is zero, and its sensitiveness is infinite; on account of frictional resistances it is necessary in practice to retain some amount of stability. The effect of friction may be regarded as the addition to the controlling force of a force f when the balls are on the point of moving outwards, and a deduction therefrom of f when they are about to move in; hence, as $\omega = \sqrt{\frac{F}{mr}}$, we may write:

$$\omega + \Delta\omega = \sqrt{\frac{F + f}{mr}}$$

$$\omega - \Delta'\omega = \sqrt{\frac{F - f}{mr}}$$

so that, through friction alone, the speed may range from $\Delta'\omega$ below ω to $\Delta\omega$ above ω, without any motion of the balls occurring. Now in an actual governor $\frac{f}{F}$ is, of course, small, and hence $\frac{\Delta\omega}{\omega}$ is small, and $\Delta\omega$ and $\Delta'\omega$ may be regarded as sensibly equal; moreover $\frac{\Delta\omega}{\omega} = \sqrt{1 + \frac{f}{F}} - 1$, whence, as $\frac{f}{F}$ is small, we have approximately:

$$\frac{\Delta\omega}{\omega} = \frac{1}{2}\frac{f}{F} \qquad (37)$$

The force f is made up of the frictional resistance of the sleeve, joints of all the link-work, and of the spindle and valve by which the engine is ultimately controlled; its absolute value depends on the mode by which governing is effected; thus in the hit-or-miss method the governor does not act directly on any valve, and hence in this case f may be kept very small, whereas in quantity or quality governing its value is necessarily increased.

As $\frac{f}{F}$ must be kept small in order to ensure efficient governing, when f is large F must also be proportionately large, i.e. a 'powerful'

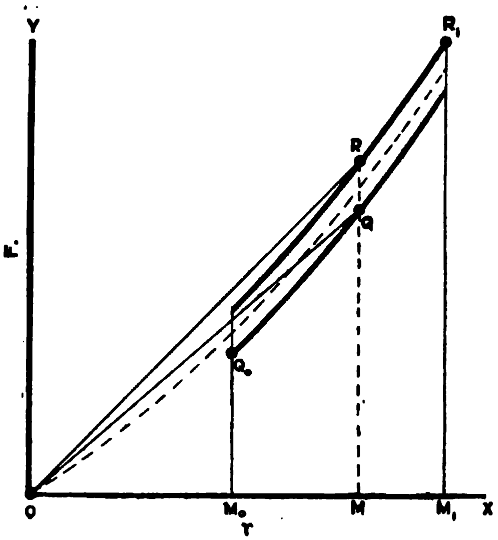

FIG. 258

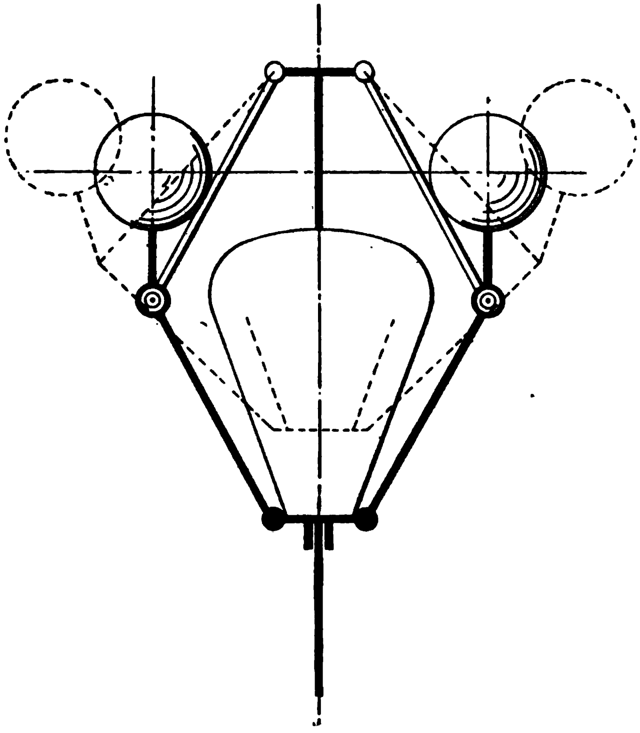

FIG. 259

governor must be used, and this is obtained by loading either by dead weight or spring or both, and using small masses revolving at a high speed. The effect of friction is shown in the curve of controlling force as indicated in fig. 258, where the central dotted curve is that of controlling force without friction and the upper and lower full line curves correspond respectively to $F + f$ and $F - f$. For any value of r, as OM, we have $F - f = MQ$ and $F + f = MR$; and at this value of r the speed may change from $\sqrt{\frac{\tan XOQ}{m}}$ to $\sqrt{\frac{\tan XOR}{m}}$, the balls at the lower speed being about to move in, and at the higher about to move out. The lowest speed possible is now that corresponding to the angle XOQ_0, and the highest that corresponding to XOR_1.

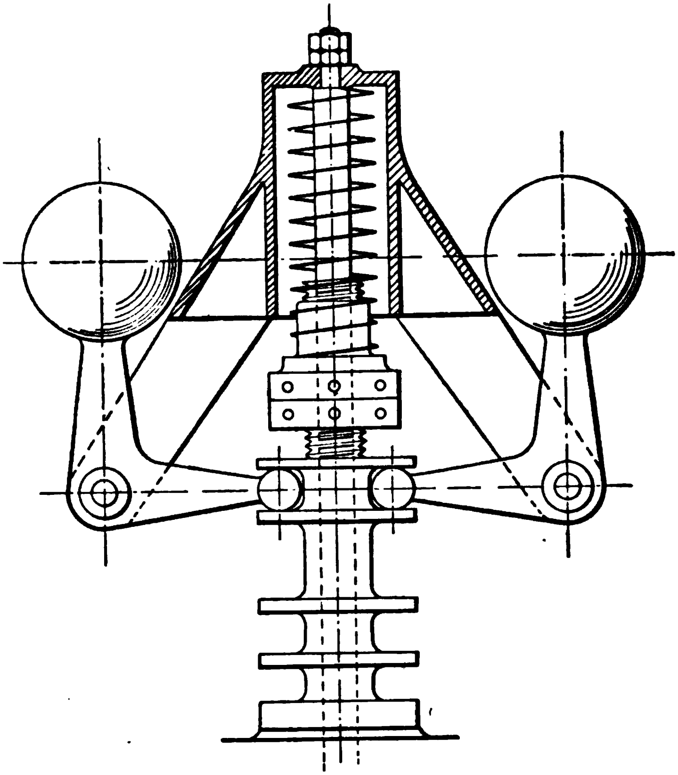

FIG. 260

Dr. Pröll's governor is an instance of a loaded dead weight ball governor which is approximately isochronous; the balls are here carried not at the junction of the links, but at the extremities of short arms standing vertically and forming part of the lower links (fig. 259); by suitably choosing the proportions of the arms and links the controlling force may be caused to vary as nearly as desired with the radius at which the balls revolve.

The Hartnell, fig. 260, is an example of a spring-loaded inverted-ball governor in which the balls move in a nearly horizontal direction, thus practically eliminating the effect of gravity. This governor is rendered isochronous by giving the spring a compression, P_0, at the minimum radius r_0 such that when the radius has become r, P_0 has increased to a value P, given by the relation $\frac{P}{P_0} = \frac{r}{r_0}$.

Ball governors, whether weight-loaded or not, are often indirectly loaded by a spring attached to some point in the link-work connecting the governor with the engine valve; the tension or compression of this spring can then be readily adjusted while the engine is running. Further, by so attaching the spring as to cause the controlling force to vary more nearly in proportion to the radius at which the balls revolve, the isochronism of the governor may be increased.

In Germany, especially during the past twenty years, the manufacture of very large gas engines and the necessities of close speed regulation have led to a considerable amount of attention being devoted to the subject of governors, and there are even some firms who specialise in governor design and manufacture; before referring to the better known of these it is desirable to give some definitions.

(1) *The Coefficient of Instability* is the value of the ratio of the difference of the maximum and minimum revolution rate of the governor to its mean revolution rate when it is not connected to the link gear operating the engine valve. Thus, if

n_1 = Revolution rate with sleeve fully raised;
n_0 = Revolution rate with sleeve at bottom;
n = Mean revolution rate;

then, very approximately, $n = \frac{n_1 + n_0}{2}$, and thus

$$\text{Coeff. of instability} = \frac{n_1 - n_0}{n} = \delta \tag{38}$$

(2) *The Coefficient of Insensitiveness.*—When the governor sleeve is connected up with the link-work gear through which the engine valve is operated, the resistance to be overcome by the governor in raising or lowering the sleeve is increased. This resistance is made up of:

(*a*) The internal friction of the governor itself.

(*b*) The resistances of the governing engine valve and the friction at the joints of the connecting link-work.

Before the governor can cause any motion in the sleeve, its speed must be increased in order to enable it to overcome this resistance, and some of its energy is thus absorbed. The coefficient of insensitiveness is defined as the ratio of the difference between the revolution rates at which the sleeve is on the point of rising and of falling, respectively, to the mean revolution rate between these; i.e. denoting this coefficient by ϵ, we have (*v.* Eq. (37)):

$$\epsilon = \frac{(\omega + \Delta\omega) - (\omega - \Delta'\omega)}{\omega} = \frac{2\Delta\omega}{\omega} = \frac{f}{F} \tag{39}$$

(3) *Force of the Governor.*—By the Force of the governor is meant the force in lbs. weight which the governor, when at rest in any position, exerts on the sleeve. In most governors this force varies for different degrees of sleeve lift; in the Hartung and Tolle designs (*v. infra*), however, the force is practically constant. The average value of the 'Force' of the governor, in this sense, will be denoted by S.

(4) *The Power of the Governor* is the product of the mean Force S and of the sleeve lift l. Denoting, then, the power by V, we have:

$$V = Sl \tag{40}$$

(5) *The Displacing Force* V is that which a governor having a coefficient of insensitiveness ϵ is able to exert, and is given by:

$$V = S\epsilon \tag{41}$$

Of the total resistance to the motion of the sleeve, the internal friction of the governor alone, G, absorbs from 3 per cent. to less than 1 per cent. of S in different well-known designs, and may be taken at the mean value of

$$G = 0{\cdot}013\,S \tag{42}$$

If the resistance of the connecting link-work and engine valve, measured at the governor sleeve, be expressed by W, then the size of the governor must be such that

$$V = G + W \tag{43}$$

As by (41) $V = S\epsilon$, we have also

$$S = \frac{G + W}{\epsilon} \tag{44}$$

Now, it is common—in order to prevent ‘hunting’—to take the coefficient of instability, δ, as 1·2 times that of fluctuation of speed as regulated by the flywheel; i.e. (*v.* p. 349), one may write:

$$\delta = \frac{1\cdot2}{m} \tag{45}$$

Moreover, it is also usual, from experience, to take $\epsilon = 1\cdot58$; hence from (45) $\epsilon = \frac{1\cdot8}{m}$; and thus Eq. (44) gives

$$\mathrm{S} = \frac{m(\mathrm{G} + \mathrm{W})}{1\cdot8} \tag{46}$$

From which the force of the required governor may be inferred from the Coefficient of Fluctuation of Speed and total resistance opposed to the movement of the governor sleeve.

In the Tables of Data published in Germany relative to governors one commonly finds information given as to the value of the displacing force corresponding to a 2 per cent. change from the normal in the revolution rate; it should be noted that in such cases the coefficient of insensitiveness ϵ is 4 per cent.; see Eq. (39).

In the design of R. Trenck, of Erfurt, shown in fig. 261, the governor is controlled partly by gravity, partly by loading, but mainly by the central helical spring. As the speed increases the balls move outward, rising slightly against gravity, and at the same time lifting the metal cover, spring case, and sleeve, and compressing the central spring. In the following short table some data (from H. Haeder) are given for the standard sizes of this governor:

Data relating to the Trenck Governor

Size No.	0	1	2	3	4	5	6	7	8
Revs. per min. . .	300	280	260	240	220	200	180	160	160
Sleeve lift, *l*, ins. .	1·18	1·57	1·97	2·36	2·76	3·15	3·74	4·53	4·53
Mean force, S, lbs. wt.	92·5	139	220	357	530	750	1000	1350	1675
Power, V, inch-lbs. .	109	218	433	840	1460	2360	3740	6125	7600
Dimension, D, ins. .	11·8	14·4	16·9	20·1	23·6	27·6	32·3	37·5	39·0
Dimension, H, ins. .	14·0	16·8	19·3	22·5	26.4	30·7	35·8	42·3	42·3

It will be noted that the pin joints in this design are subjected to the action of considerable forces when the governor is at work; special provision is made for lubricating both these joints and the sleeve, as will be seen on reference to fig. 261, in order to reduce friction as much as possible; the governor can only be oiled, however, when it is at rest.

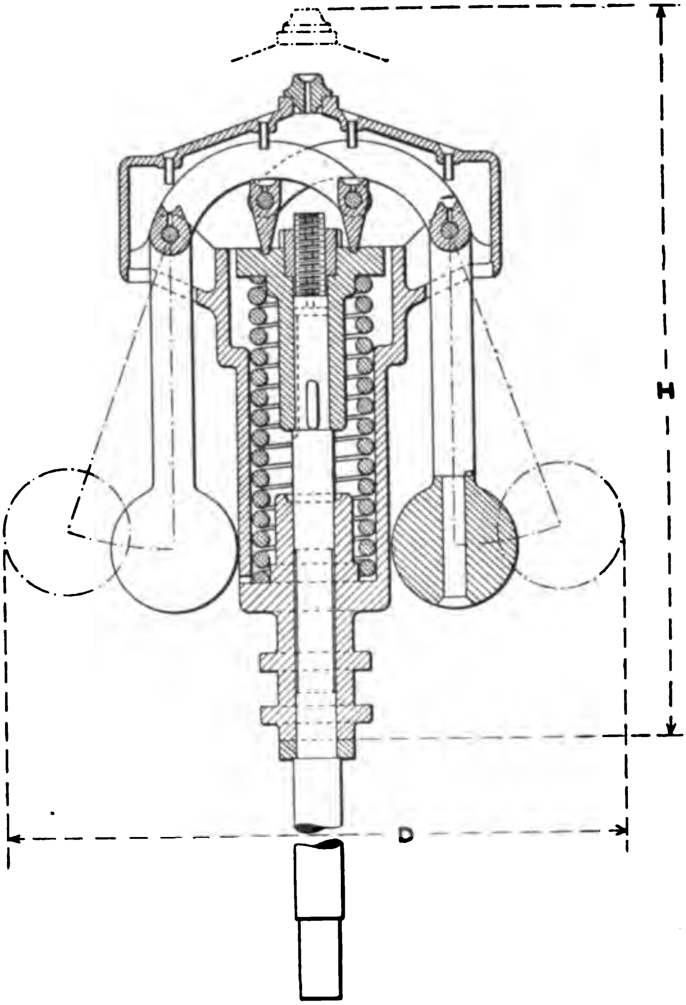

FIG. 261

In fig. 262 the spring-loaded governor of F. Beyer & Co., Erfurt, is illustrated in section. The balls are here inverted and, descending slightly as they move outward, are to a slight extent assisted by

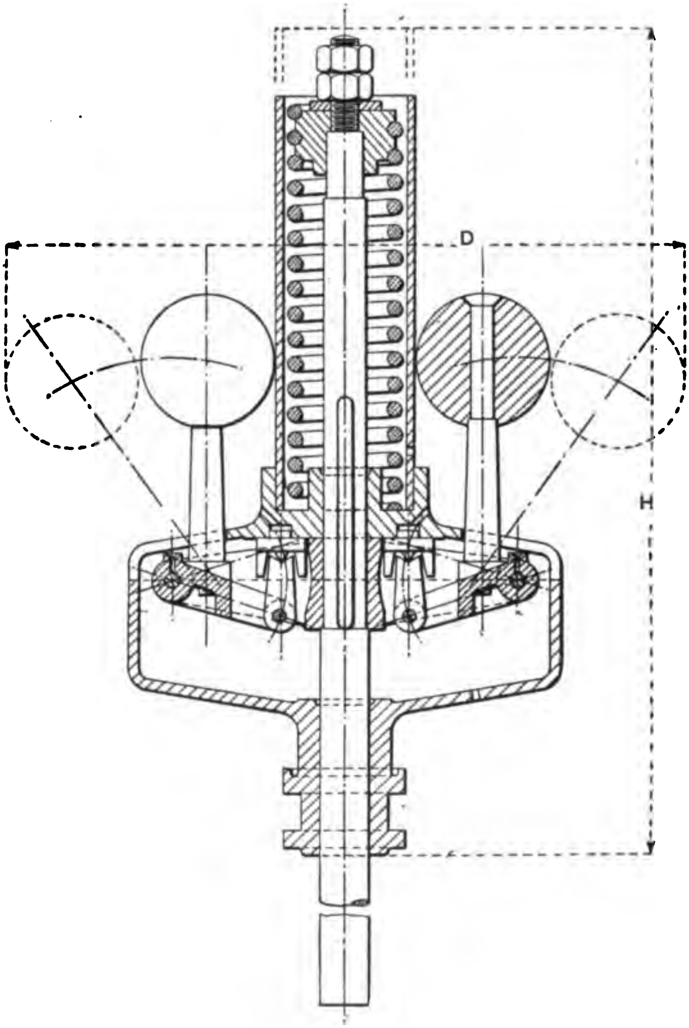

FIG. 262

gravity. The control is by the long helical spring as shown. Here also the pin joints are subjected to considerable forces during working, and lubrication can only be effected when the governor is at rest.

The range in size and capacity may be seen from the following table:

DATA RELATING TO THE BEYER GOVERNOR. (*From Haeder*)

Size No.	10	11	12	13	14	15	16	17
Revs. per minute.	300	280	260	240	220	200	180	160
Sleeve lift, *l*, ins.	1·18	1·57	1·97	2·36	2·76	3·15	3·74	4·53
Mean force, s, lbs. wt. . .	93·5	192	330	520	740	1045	1490	2100
Power, v, inch-lbs.	110	301	648	1227	2040	3300	5570	9530
Dimension, D, ins.	12·2	15·5	19·3	22·4	25·8	30·0	34·9	40·7
„ H, ins.	15·5	20·0	23·2	26·0	30·5	35·5	40·0	45·7

In the well-known governor of Messrs. Hartung, Kuhn & Co., Düsseldorf, which appeared about 1893, the use of balls is abandoned, and friction on the pin joints almost wholly eliminated by adopting the general arrangement shown in sectional elevation and plan in fig. 263.

The centrifugal masses and springs here interact without any intervening mechanism, the spring compression proceeding *pari passu* with the outward movement of the blocks. The joints of the bell-crank levers are subjected only to forces arising from the resistance at the sleeve, and the friction of the governor itself is accordingly very small; the springs can also evidently be so adjusted as to initial compression that the action of the apparatus may be made as nearly isochronous as desired.

Some data relating to standard sizes of Hartung governors, compiled from H. Haeder's treatise, are given in the subjoined table:

DATA RELATING TO THE HARTUNG GOVERNOR

Size No.	91	92	93	94	95	96	97	98	99	100	101	102
Revs. per minute .	340	310	240	240	210	200	190	180	165	160	140	130
Sleeve lift, ins. .	0·79	1·0	1·18	1·18	1·58	1·97	2·36	2·76	3·15	3·54	3·94	4·33
Mean force, s, in lbs.	125	180	250	330	415	525	580	660	880	1325	1650	2200
Power, v, inch-lbs.	98·5	180	295	390	655	1035	1370	1820	2770	4690	6500	9550
Dimension, D, ins.	10·4	12·2	13.8	15·0	16·5	18·1	19·7	21·6	26·0	30·7	33·9	37·8
„ H, ins.	10·9	11·8	14·2	15·3	16·5	17·3	19·1	20·1	22·2	24·3	27·2	28·8

In the later Steinle-Hartung design, of which a section is shown in fig. 264, the disposition of the levers by which motion is communicated from the centrifugal masses to the sleeve is altered, but otherwise the general arrangement is much as before; this design is manufactured by Messrs. Steinle and Hartung of Quedlinbourg.

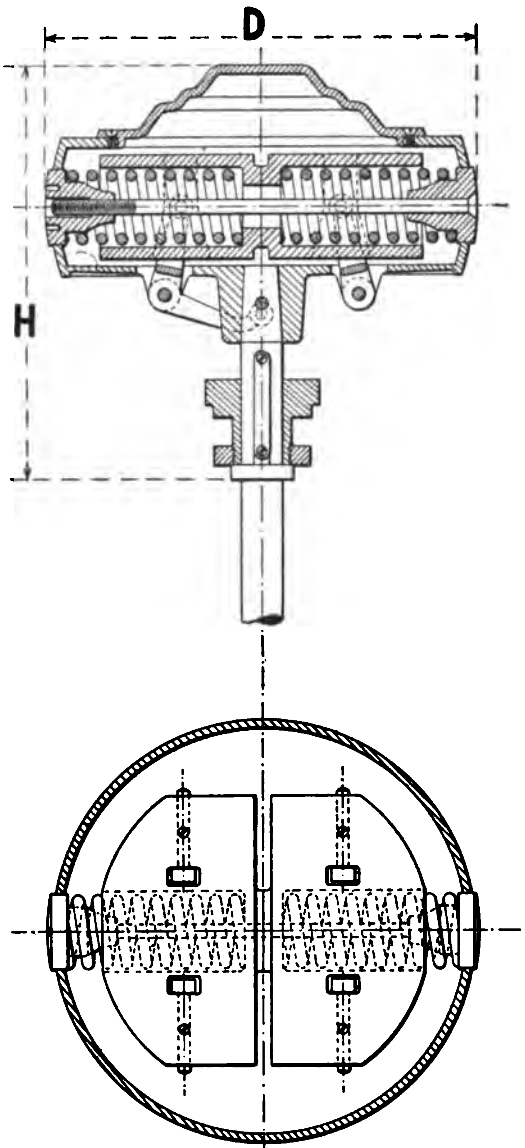

FIG. 263

Both in the Hartung and Steinle-Hartung designs the lower flange of the sleeve is screwed on; this allows a trunnion ring to be used in the groove instead of the usual trunnion blocks, which is an improvement, inasmuch as greater bearing surface is provided, and wear can be easily taken up, thus preventing the lost motion so common after use here in the ordinary type. The following table gives some particulars of the standard sizes of Steinle-Hartung governors:

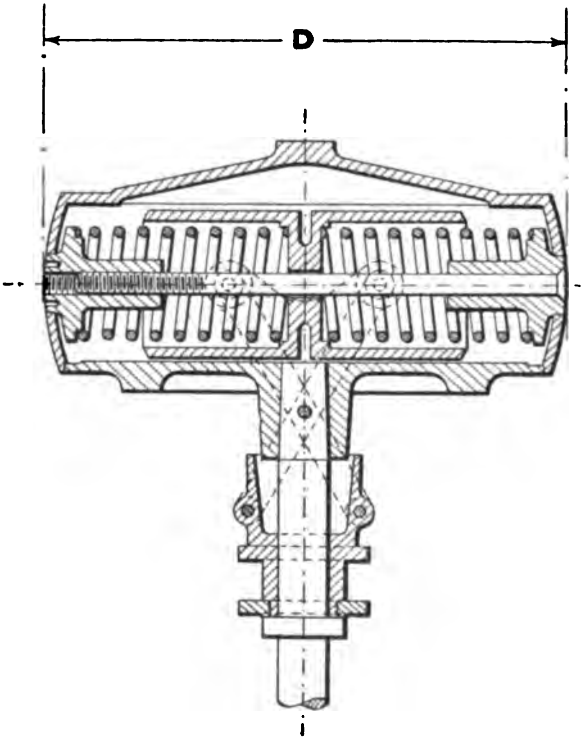

FIG. 264

DATA RELATING TO THE STEINLE-HARTUNG GOVERNOR. (*Haeder*)

Size No.	92	93	94	95	96	97	98	99	100
Revs. per minute .	320	300	280	270	260	250	240	220	210
Sleeve lift, *l*, ins. .	1·0	1·18	1·58	1·97	2·36	2·75	3·15	3·54	3·94
Mean force, s, lbs. wt.	88	176	300	403	550	730	960	1320	1650
Power, V, inch-lbs. .	88	208	475	475	1300	2010	3020	4670	6500
Dimension, D, ins. .	10·2	12·6	14·6	16·2	17·7	19·7	22·1	24·8	27·6

As already remarked, in the Hartung governors the force, S, is practically constant. The displacing force V, in lbs. weight (*v.* Eq. (41)), exerted at the sleeve in the Steinle-Hartung governors for a 2 per cent. variation of speed from the normal (so that $\epsilon = \frac{1}{25}$), and the corresponding 'useful power,' Vl, in inch-lbs. are as follows :

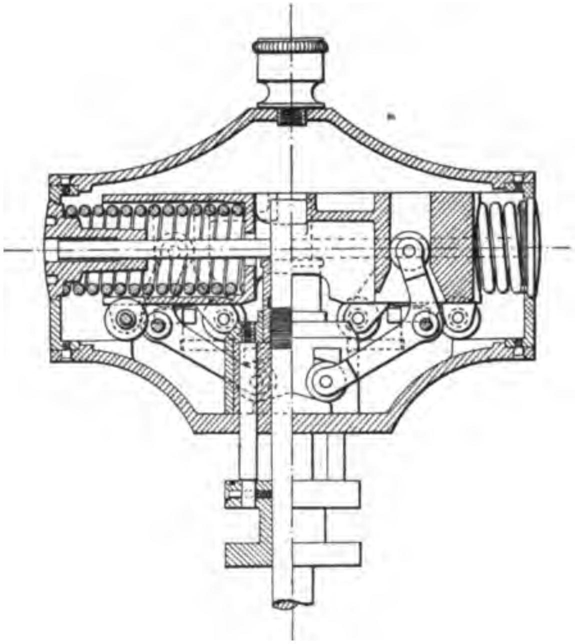

FIG. 265

Size No.	92	93	94	95	96	97	98	99	100
V in lbs. wt. .	3·52	7·04	11·9	16·1	22·0	29·3	38·5	52·8	66
Vl, inch-lbs. .	3·52	8·3	18·8	31·7	51·9	80·7	121·0	187·0	260

In the Hartung and Steinle-Hartung governors the weight of each centrifugal mass is borne by the pin joint in the upper end of the bell-crank. In the Jahns governor, figs. 265 and 266, these masses are carried each upon three rollers running upon ways formed in the bottom of the casing; the masses thus move in a direction always strictly horizontal. The upper end of each bell-crank is fitted with a roller and moves slightly up and down in the slotted way provided, as indicated in the sectional view. Each mass is further fitted at each lower outer corner with a roller revolving in a horizontal plane; these prevent any side motion and greatly reduce the side friction of the masses.

In the Jahns governor it will be noted that the casing is entirely enclosed, and is thus grit and moisture excluding, and the moving parts work in an oil bath.

The design of Messrs. Rost of Dresden (fig. 267) is unique, inasmuch as the two centrifugal cylindrical masses are suspended, and mainly controlled from the free ends of a pair of spiral springs. These springs are attached to square arbors, and the controlling force can be varied by means of the adjusting screw A, acting simultâneously and equally upon both. The following figures (from Haeder) are useful for reference:

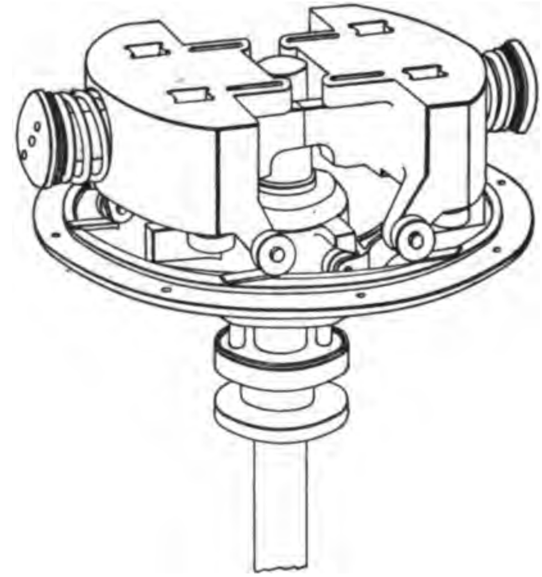

FIG. 266

DATA RELATING TO THE ROST GOVERNOR

Size No.	1	2	3	4	5	6
Revs. per minute	320	320	320	280	240	200
Sleeve lift, *l*, ins.	1·34	1·54	1·81	2·21	2·68	3·35
Mean force, s, lbs. weight . . .	143	253	473	815	1045	1600
Power, v, inch-lbs.	192	390	855	1800	2800	5350
Dimension, D, ins.	11·8	13·6	15·8	19·3	23·2	28·8
„ H, ins.	16·3	18·9	22·1	27·0	32·3	40·2

Among other well-known modern governors may be mentioned the Proëll and Tolle designs; the Tolle is manufactured by Messrs. T. Wiedes, of Chemnitz; introduced in 1895, it generally resembles the Hartung, but uses one spring under tension in place of the two under compression employed in the latter type. The Tolle governor is made in eight standard sizes, running at speeds ranging from 400 revolutions per minute in the smallest to 270 revolutions per minute in the largest pattern; it is thus a high-speed governor.

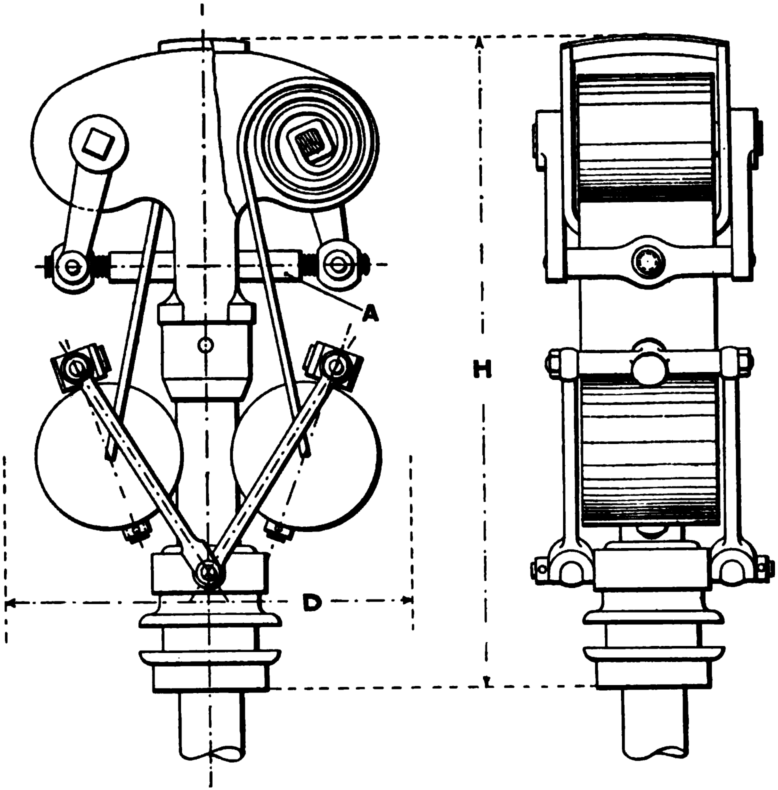

FIG. 267

A sufficient account of modern governors has, it is hoped, been given above to enable the reader to appreciate the general trend of design and the amount of care and attention bestowed in recent years upon this important accessory.

We conclude with a brief reference to some typical governing arrangements adopted in modern large gas engines. We remark that in recent years it is noticeable that several important firms have at one time used one system of governing and at another a different system; hence it may be inferred that there is no one method which is best in all cases. Reference has already been made to the National Gas Engine Co.'s design in which the governing is successively by quality, quantity, and hit-or-miss, all obtained by one simple arrangement of mechanism. The author, in his Cantor Lectures, 1905, cited experiments made by Mr. Bradley of the National Co. upon a 9 in. × 17 in. engine using (*a*) hit-or-miss; (*b*) quantity; and (*c*) quality, governing; the results are exhibited in the accompanying table:

MR. BRADLEY'S GOVERNING TESTS WITH A 9" × 17" NATIONAL GAS ENGINE AT 200 REVOLUTIONS PER MINUTE

BHP	Method of Governing		
	Hit-or-miss Gas per BHP hour	Quantity Gas per BHP hour	Quality Gas per BHP hour
	cub. ft.	cub. ft.	cub. ft.
6·75	22·4	—	—
6·85	—	24·2	36·4
8·44	19·1	—	—
8·75	—	20·4	29·1
10·3	18·2	—	—
10·4	—	—	25·1
10·5	—	19·9	—
18·1	—	—	17·7
18·2	16·75	17·5	—

These figures show that the best economy of gas was obtained with hit-or-miss governing, and that the quality method gave the worst results; governing by variable quantity, i.e. by throttling a uniform mixture, furnished consumption figures somewhat in excess of the hit-or-miss method throughout, but on account of the greater uniformity in the working impulse obtained with the quantity method, it is extensively employed, especially with large engines.

The Cockerill Co., of Seraing, have standard designs for both quality and quantity governing, and use the one or the other method according to the nature of the work to be done by the engine. One of their arrangements for quality governing is illustrated in fig. 268. The mixture inlet valve is opened by the usual cam, roller, rod, and lever G, and is closed by the stiff helical spring shown at the top of the spindle; fixed also on this spindle, above the valve, is a cylindrical slide by which air is admitted throughout the suction stroke. The double-seated equilibrium gas valve is guided by—but moves independently of—the inlet valve spindle, as shown; a helical spring is in

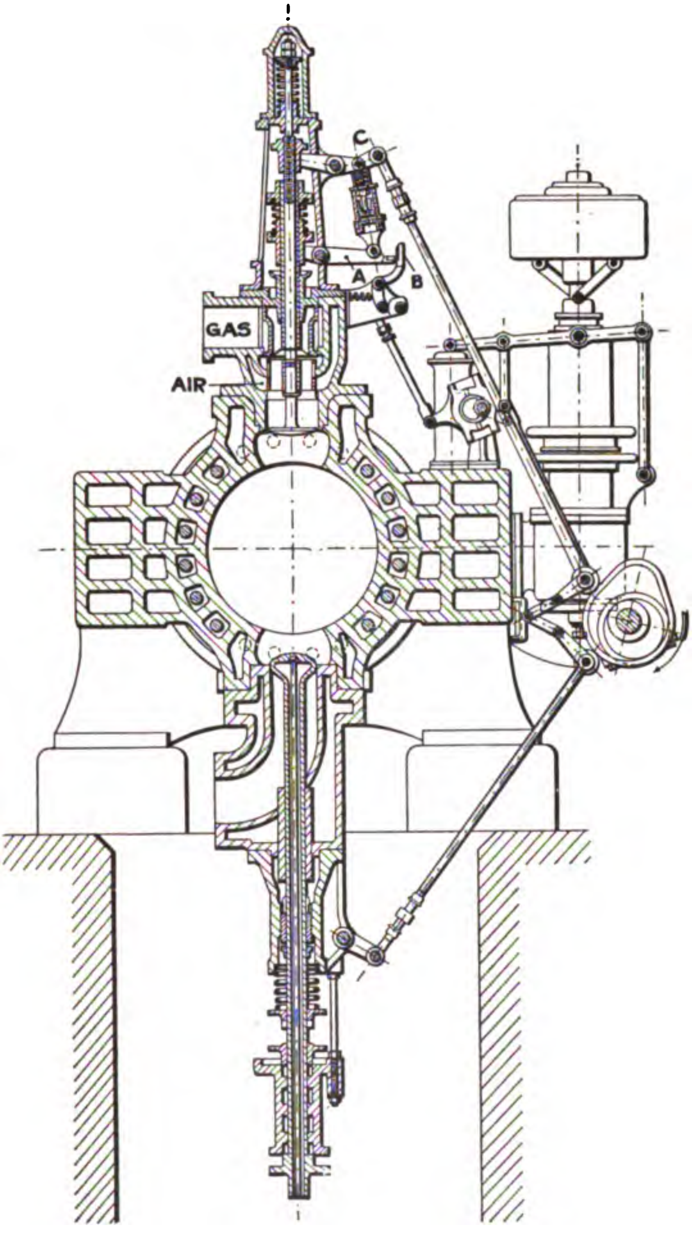

FIG. 268

compression between the top of the gas valve guide and an adjustable collar on the inlet valve spindle. The gas valve is held on its seat by the lever A, fulcrumed in the valve casing wall, its outer end being held down by the trip lever B controlled by the governor. The lever A is also connected, by the small air-cylinder and piston shown, to the lever C, by which the inlet valve is lifted.

The inlet valve cam opens the valve and attached slide through the lever C, thus admitting air only throughout the suction stroke; as the inlet opens, the helical spring on top of the gas valve guide is compressed; this valve cannot at first open, as it is held on its seat by the trip B being engaged with the outer end of the lever A.

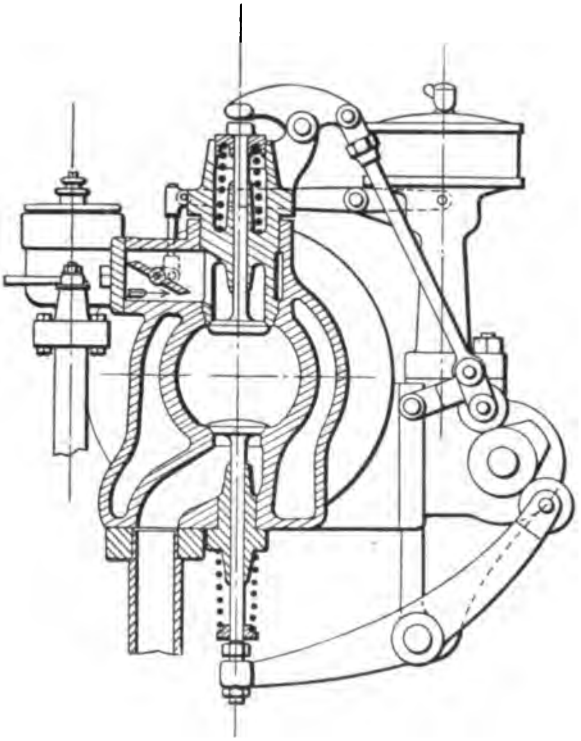

FIG. 269

At a determined point in the suction stroke, however, the governor moves the trip lever B, thus releasing A, and the gas valve is then suddenly opened by its compressed spring; gas thus enters during the latter portion of the suction, and a rich and readily ignitible mixture is thus ensured in the neighbourhood of the igniters on compression.

The descent of C when the inlet closes causes the gas valve also to close through the action of the air cylinder connecting C and A, and the trip lever then again engages the free end of A in readiness for the next admission. In the variable quantity arrangement of the Cockerill Co. the mixture inlet valve is of constant stroke and, as before, is actuated by a cam and closed by a stiff helical spring. To a sleeve sliding on the spindle of this valve are attached a cylindrical air regulator and a double-seated equilibrium gas valve, which in this case move together, their lift being controlled by the governor, which 'cuts off' the mixture sooner or later in the suction stroke according to the speed of the engine.

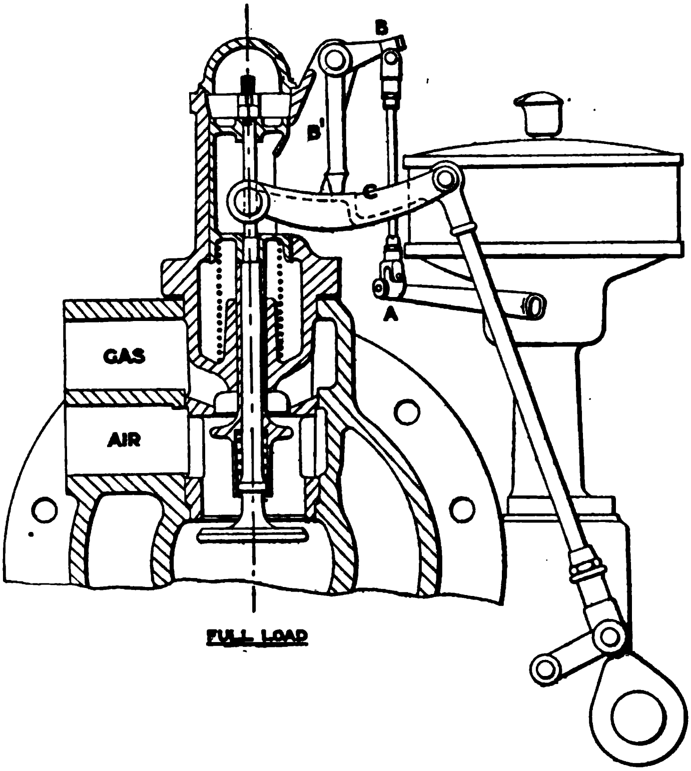

FIG. 270

The Cockerill Co. appear on the whole to favour quality governing, and in their later designs have also somewhat simplified the arrangement illustrated in fig. 268 by substituting a simple disc valve for the double-seated equilibrium gas valve there shown.

The simple quantity-governing arrangement of the smaller horizontal gas engines of Messrs. Crossley Bros. is illustrated in fig. 269; the inlet valve lift is constant, while the quantity of mixture admitted through it is regulated by the governor acting through the throttle valve fitted in the inlet port as shown.

In the Otto engines of the Deutz Co., and more recently in those of the Crossley Co., especially in the larger sizes, governing is by variable quantity by the now well-known device of a governor-controlled fulcrum to the inlet valve lever, by which the lift of the valve is varied while the duration of opening is not affected. Messrs. Crossley's arrangement is diagrammatically illustrated in fig. 270. The end A of the governor lever is connected by a link with the bell-crank B B′, the free end of which forms the fulcrum about which the inlet valve lever C rocks.

The lever C is operated through the usual cam, roller, and rod, as shown; thus the duration of opening of the inlet valve is constant, while the amount of its lift is dependent upon the position of the fulcrum arm B′.

When the speed of the engine increases, the end of the governor lever A descends, depressing B, and consequently moving B′ towards the left; the lift of the inlet is thus reduced, and the charge consequently throttled throughout admission.

In fig. 270 it will be seen that the gas valve is a simple disc valve carried on the inlet valve spindle, and that there is no valve specially controlling the admission of air before it passes the main inlet valve; also that the main inlet and gas valves are seated by the action of the stiff helical springs under compression, as indicated. In fig. 268 there is a cylindrical sliding valve regulating the air admission, and the gas valve is of the double-seated equilibrium type; this latter valve, as already mentioned, has been replaced by a simple disc in the more recent practice of the Cockerill Co. In the double-acting engines of the Otto Co. of Deutz the practice of placing the gas and air valves on the main inlet valve spindle has been discarded, the view being taken that this very usual arrangement involves unnecessarily heavy working parts in large engines, and further that, as the combined gas and air valve is most liable to get fouled up in working, it is best contained in a separate and easily accessible valve-box. In their horizontal double-acting 2000 HP type engine, for example, the gas and air valves are in a valve-box located midway between the two main inlet valves on the top of the cylinder, and the admission of gas and air is varied as to quantity by varying the lift of these valves simultaneously by means of a governor-shifted fulcrum as already described. Moreover, one combined gas-and-air valve thus located supplies both the main inlet valves of the cylinder, which effects an important saving in construction and ensures also identity of mixture quality and quantity at each end of the cylinder.

Placing the gas and air valves in a separate valve-box involves, however, an increased volume of port between these and the main inlet valves, and this volume is filled with fresh mixture, and hence

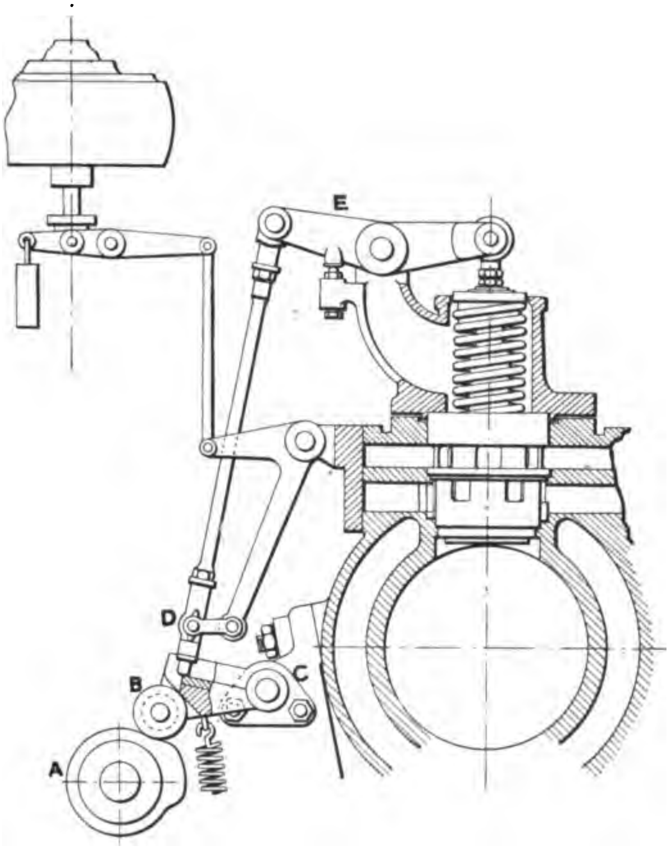

FIG. 271

would tend to prevent the governor from controlling the engine speed quite as closely and promptly as in the more usual arrangement, wherein the gas and air valves are fitted on the spindle of the main inlet valve.

In the earlier Nürnberg engines the valve lift was controlled by the governor by the simple linkage arrangement illustrated in fig. 271. The lever B, hinged at C, is actuated by the cam A. The lower end of the rod D makes contact with B at a greater or lesser distance from its fulcrum C determined by the governor through the connecting link-work shown. The lift of the inlet valve is of constant duration, and is caused by the cam A through the rod D and rocking lever E, while the amount of the lift is controlled by the governor.

The later Nürnberg arrangement, with which that of Schmitz is also practically identical, is illustrated in fig. 272, which shows the arrangement of linkage whereby the governor determines the position of the roller end of the lever C between the surfaces A and B, and thus regulates the lift of the inlet valve operated by the cam D. In the position shown the lift is at its maximum value, while when in the configuration indicated by the dotted lines it is reduced almost to zero.

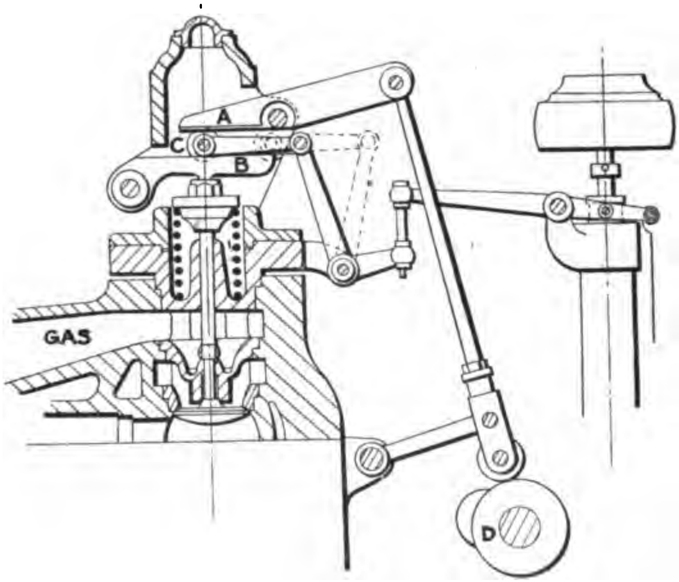

FIG. 272

In the Tangye & Robson arrangement, fig. 273, the cam A actuates the rocking lever B, which gives motion to the inlet valve through the roller C and end D of the lever, hinged at E to the link F. The governor, operating through the link-work shown, varies the position of the roller C relatively to the fulcrum of the rocking lever B, and so varies the lift of the inlet valve. Note that the duration of lift is constant the amount only being changed, so that the governing is by quantity, admission being throttled throughout the suction stroke.

The early gas engines of Ehrhardt & Sehmer were constructed under licence of the Otto Co. of Deutz, and included the usual Deutz shifting-fulcrum quantity governing arrangement already described, in which the mixture at light loads is throttled throughout the suction stroke.

In their later practice the Deutz system has been discarded, and a sectional view of the arrangement adopted is given in fig. 274. There is no valve regulating the admission of air, but instead a regulating disc, A, adjustable by hand, by which the proportion of air to gas may be varied; this disc acts also as a baffle, and deflects the gas and air streams so as to prevent them from impinging directly on one another.

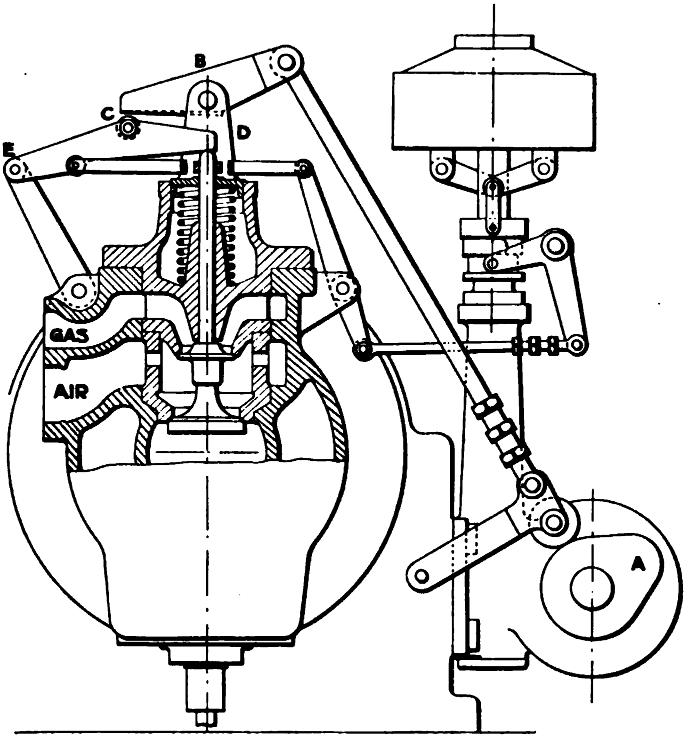

FIG. 273

The gas valve is of the double-seated equilibrium type, and is raised by an eccentric acting through a tension rod B, trip C, and rocking lever D.

The gas is admitted from the beginning of the suction stroke, and towards its end, at a point depending upon the engine speed, it is cut off by the governor releasing the trip through the link-work shown ; the gas valve, on release of the trip, is quietly closed by the compressed helical spring contained in the casing at the top of the device, which is fitted to act also as an air-cushioning ' dash-pot.'

Air alone continues to enter the cylinder during the remainder of the suction stroke, and thus completely scavenges the port between the gas valve and main inlet valve, thus preventing any chance of back-firing. The governing is thus by variable quality, with the peculiarity that the gas, contrary to the usual practice, is always admitted at the beginning of the suction stroke, and cut off before its end, whence it might be inferred that a poor mixture or even comparatively pure air would alone remain in the neighbourhood of the igniters. Owing, however, to the turbulence of the entering mixture, this does not appear to occur, and it is said that the firing is regular and brisk under all circumstances.

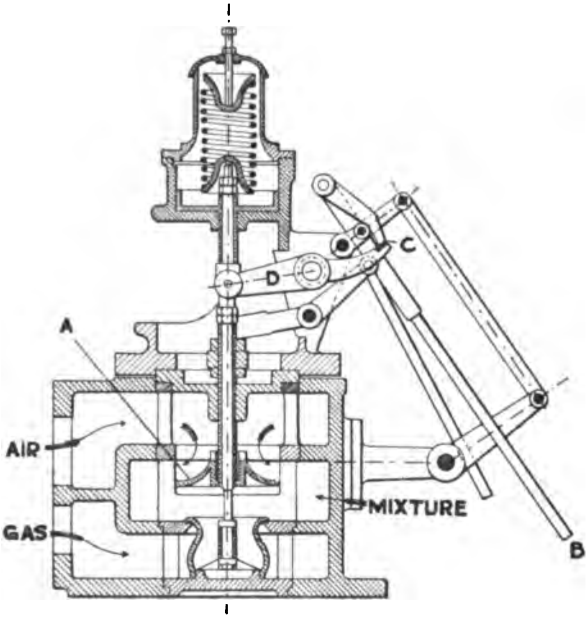

FIG. 274

Mr. R. E. Mathot, to whose exhaustive critical examination of the governing systems of European gas engines [1] the authors are indebted, points out that for large central stations in cases where the power gas is not absolutely clean, the Nürnberg practice of separating the gas and air valves from the main inlet is likely to be useful, as these valves are then more readily accessible for examination and cleaning; on the other hand, as already mentioned, the increased volume of port between the mixing valves and the main inlet may adversely affect the closeness of the speed regulation by the governor. Mr. Mathot suggests that from the points of view both of design and working the type in which the air and gas valves are carried on the main inlet valve spindle, all three being thus contained in one common casing with a minimum volume of mixing chamber, is likely to become standard practice. He inclines also to the view that the double-seated balanced gas valve furnishes better mixing than the simple disc valve, and points out that sliding air valves are open to the objection that they require regular lubrication, which assists the deposition of dust and suspended matters in the air, and thus in the end defeat their own object. Trip gears are now much used in the large gas engines of the principal European builders, especially when of the double-acting type, to which the ordinary Deutz shifting-fulcrum design is not immediately applicable on account of the fulcrum lever never becoming free in such cases, and the governor action being consequently hampered; this difficulty has, however, now been removed by the ingenuity of the Deutz designers, who have satisfactorily applied their standard method of governing to large double-acting engines, as already stated.

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[1] *Gas and Oil Power*, vols. iii. and iv.

CHAPTER V

GASEOUS FUELS FOR INTERNAL COMBUSTION ENGINES

The principal gaseous fuels now used in internal combustion engines are town gas, producer gas, coke oven gas, and blast furnace gas. Of these the most important in the earlier stages of the development of the gas engine was town illuminating gas, and town gas is still the practically exclusive fuel used for the smaller engines, up to about 30 BHP. Town gas is quite suitable from the technical point of view for the largest gas engine, and indeed it is now used at the Crystal Palace for actuating two engines each of 1000 BHP and one of 750 BHP, driving dynamos for supplying the electric light required. The only limitation on the use of town gas for large powers is due to its relatively high cost per heat unit generated. If coal gas could be distributed to the consumer at a sufficiently low price, no other motive power whatever would be used in our towns. Indeed, if coal gas could be generated at the pit's mouth and distributed under moderate pressures to the nearest cities for power, heating, and lighting, great economic advantages would ensue by the savings which could be effected in fuel and heat consumption. Gas engine inventors and designers are greatly indebted to the town gas industry; without coal gas the internal combustion engine could not have attained its present position.

Accordingly it is fitting that town gas should be first considered.

Town Gas.—The engineers of the coal gas industry have fully recognised the changed conditions introduced by the growth of the gas engine, the invention of the incandescent gas light, and the extension of the consumption of gas for cooking and heating, and they are endeavouring to produce a gas which can be distributed to the consumer at the lowest price per heat unit available. Broadly, the internal combustion engine requires a clean, pure gas which evolves the highest possible number of heat units at a given cost; flame temperature need not be high nor need heating value per cubic foot, so long as the price per heat unit is low. What is true for the gas engine is also true for gas cooking and heating. The incandescent gas light, however, requires high flame temperature; 2000° C is desirable.

Accordingly, to meet all these conditions it is necessary to have a gas giving about 500 B.Th.U. per cub. ft. as its lower calorific or heating value.

In the early days it was necessary for the gas to have a high intrinsic illuminating power, and Cannel coal was mainly depended on to supply the heavy hydrocarbons required. This necessitated increased cost to maintain the gas at the point required by law. In recent years Parliament has consented to a reduction in intrinsic illuminating power, and accordingly a cheaper gas per heat unit evolved has been distributed to the public, greatly to their advantage both in industry and at home.

It is to be hoped that the movement will continue until most towns in the country have a supply of pure gas of about 500 B.Th.U. lower heating value at from 1*s.* to 1*s.* 6*d.* per thousand cub. ft.

A change of chemical composition of the gas has accompanied the fall of intrinsic illuminating power, as may be readily seen by the comparison of the analyses of London gas in 1884 and in 1909, given below.

COMPOSITION OF LONDON ILLUMINATING GAS BY VOLUME

—	1884		1909		
	Coal gas	Coal gas	Coal gas	Mixed gas	Carburetted water gas
Hydrogen, H	47·99	53·14	49·21	46·73	29·35
Methane, CH$_4$	37·64	36·55	31·22	24·57	20·48
Carbonic Oxide, CO	3·75	4·11	8·17	14·46	33·19
Unsaturated hydro-carbons	4·41	2·92	3·39	4·91	11·32
Carbon dioxide	1·50	0·09	1·48	1·45	—
Oxygen	0·26	0·26	0·26	0·73	—
Nitrogen	5·95	3·19	6·27	7·15	5·66
Lower heat value in B.Th.U. per cub. ft. at 60° F. and 14·7 lbs. pressure	584	557	514	513	598

In 1884 town gas was made by the destructive distillation of coal only, and later coal gas was mixed with enriched or carburetted water gas, which consists mainly of hydrogen and carbonic oxides produced by the action of steam on incandescent coke. Carburetted water gas is also used alone. The coal gas analyses are easily recognised by the low percentage of carbonic oxide; the mixed gas above contains 14·46 per cent. of carbonic oxide, while the carburetted water gas contains 33·19 per cent. of that gas. It will be noted that the heat value of the coal and mixed gas falls from a mean lower heat value of 570 B.Th.U. per cub. ft. at 60° F. and 14·7 lbs. per sq. in. in 1884 to 514 B.Th.U. in 1909, a fall of 10 per cent.

The heat value of the carburetted water gas is, however, 598 B.Th.U. in 1909, against the coal gas mean value of 570 at the earlier date, an increase of 5 per cent.

Mr. J. H. Coste, F.I.C., F.C.S., discusses this question in his interesting book, and comes to the conclusion that some fall in calorific power has accompanied the modern movement for cheaper gas; the following table is prepared from his calculations:

CALORIFIC POWER OF LONDON GAS AT DIFFERENT DATES

Reputed illuminating power.		Year.	
— —	Coal gas	1851	575 B.Th.U. per c. ft. lower heat value
16 candles	,, ,,	1876	575 ,, ,, ,, ,, ,,
16 ,,	,, ,,	1907–1909	527 ,, ,, ,, ,, ,,
14 ,,	Mixed gas	,, ,,	504 ,, ,, ,, ,, ,,
16 ,,	,, ,,	,, ,,	520 ,, ,, ,, ,, ,,

The 16-candle coal gas in 1909 is thus lower in calorific value than that of 1876 by 9 per cent., while the 16-candle mixed gas of 1909 is 10·5 per cent. lower, and the mixed gas of 14 candle-power is 14 per cent. lower. A weekly summary of gas testing is prepared by the superintending gas examiner for the London County Council, and Mr. Coste calculates the averages for the year 1910 for London coal gas and mixed gas as follows:

LONDON ILLUMINATING GAS, AVERAGE OF WEEKLY SUMMARY FOR 1910

London coal gas . .	583 B.Th.U. higher, and	520 B.Th.U. lower, value.
London mixed gas . .	548 ,, ,, ,,	488 ,, ,, ,,

The average lower calorific value throughout a year, per cub. ft. at the meter for coal gas is 520 B.Th.U. per cub. ft., and for mixed gas 488 B.Th.U.

Mr. Coste also shows that the same summaries prove that the South Metropolitan Company's pure coal gas varied within one week in higher calorific value from 552 to 636 B.Th.U. per cub. ft., and in lower calorific value from 493 to 570 B.Th.U.

In making calculations of cost of gas fuel in absence of direct measurement by calorimeter it is therefore advisable to take the present (1912) heating value of London coal gas and London mixed gas at 500 B.Th.U. per cub. ft. at the meter.

The price of 10,000 B.Th.U. with London gas at 2*s*. per 1000 cub. ft. is 0·48*d*., and as all makers guarantee their engines to give more than 1 BHP hour for 10,000 B.Th.U., it may be taken that the cost of gas fuel in London is less than one half-penny per BHP hour.

The lowest price of gas for power at present in the Metropolitan area is 1*s.* 9½*d.* per 1000 cub. ft. (Wandsworth and Putney), and the highest is 3*s.* 8*d.* (North Middlesex), so that even there the cost is less than 1*d.* per BHP hour.

The variations in calorific power from day to day in the author's laboratory in Holborn have been frequently noted; in the following table some results obtained in the autumn of 1910 are exhibited:

VALUES BY JUNKER CALORIMETER IN B.TH.U. PER CUB. FT., MEASURED AT WORKING TEMPERATURE AND PRESSURE

Date	Higher	Lower	Room temp., °C.	Bar., inches
28/10/11	—	504·0	18·0	29·50
30/10/11	562·0	504·5	21·0	29·50
1/11/11	590·0	532·5	21·5	29·93
2/11/11	576·0	518·0	20·5	29·70
20/11/11	538·5	483·5	17·0	29·30
22/11/11	554·5	497·2	23·5	29·26

The following table gives the composition and calorific value of town gas at recent dates:

COMPOSITION AND CALORIFIC VALUE OF DIFFERENT TOWNS' GAS

—	Hydrogen, H	Methane, CH_4	Carbonic oxide, CO	Heavy hydrocarbons	Carbon dioxide, CO_2	Oxygen	Nitrogen	Calorific value in B.Th.U per cb. ft. at 60°F and 14·7 lbs. pressure		—
								Higher	Lower	
Ashton-under-Lyne	41·29	33·73	7·13	4·74	2·62	0·27	10·22	622	561	Inst. C. E. Committee, 1905
Birmingham .	43·4	33·67	9·46	4·48	0·16	0·13	8·7	—	553	Webber, 1907
Cambridge .	47·2	35·2	7·15	4·8	—	0·25	5·4	640	585	Hopkinson 1905
London . .	45·2	29·9	10·3	5·10	—	0·5	10·0	598	538	Clerk, 1900
„ . .	42·8	27·8	11·5	5·4	—	0·1	12·0	642	577	Bairstow & Alexander
„ coal gas .	49·21	31·22	8·17	3·39	1·48	0·26	6·27	573	514	Coste, 1909
„ mixed gas	46·73	24·57	14·46	4·91	1·45	0·73	7·15	574	513	Coste, 1909
„ Gas Light & Coke Co.	29·35	20·48	33·19	11·32	—	—	—	664	598	Coste, 1909
London & S. of England .	43·38	29·29	15·62	4·64	1·5	0·26	5·31	—	540	Webber, 1907
Leeds . .	52·9	35·2	6·5	4·2	1·1	—	0·1	—	—	Groves, 1895
Coal gas: Newcastle coal	48·49	35·9	6·61	3·83	0·12	nil.	5·05	—	—	Webber, 1907
Manchester: (Openshaw).	42·9	30·93	14·99	6·8	2·14	nil.	2·24	—	578	Frankland, 1904
Sheffield. .	46·86	35·6	7·35	5·42	0·25	0·43	3·22	691	620	Thomas, 1905

The town gas of the Midlands and Northern counties and the north is richer than that of the south country. Near the colliery districts it may be taken as varying between 550 and 600 B.Th.U. lower calorific value per cub. ft. Moreover, the price is in many cases lower; thus there are thirty-three cities and towns in England where gas for motive power is sold at 1*s.* 9*d.* per 1000 cub. ft. and under; of these thirty-three cities twenty-one sell at 1*s.* 6*d.* and under, and at three places—Lancaster, Sheffield and Widnes—town gas can be had for motive power at 1*s.* per 1000 cub. ft.

Taking 550 B.Th.U. as the average lower calorific value and the cost as 1*s.* per 1000 cub. ft., we get 10,000 B.Th.U. for 0·221*d.*, and at 1*s.* 6*d.* for 0·327*d.* In three places we can thus get 1 BHP hour for less than one farthing fuel-cost for town gas; and in twenty-two places 1 BHP hour for less than one-third of a penny.

The places selling at 1*s.* 6*d.* per 1000 ft. and under are Bristol, Birmingham, Burton, Bury, Darlington, Derby, Huddersfield, Keighley, Lancaster, Leamington, Leicester, Nottingham, Oldbury, Plymouth, Sheffield, Smethwick, Stockport, Stockton-on-Tees, Walsall, Wolverhampton, and Widnes. The author gives the names, as he considers their example worthy to be followed over the whole country, tending as it does to the increased prosperity of many people and industries.

COMPOSITION OF PRODUCTS OF COMBUSTION FROM COAL GAS

The amount of air required for the complete combustion of coal gas varies with the composition of the gas. The Cambridge coal gas used by Hopkinson in his experiments on gaseous explosions, of which the analysis is given in the table, p. 411, required, as determined by experiment, 5·76 cub. ft. of air for every cub. ft. of coal gas burned.

One cub. ft. of this gas when burned with 9 cub. ft. of air produces a mixture of the following percentage composition:

	Per cent. by volume
Steam, H_2O (assumed gaseous)	13·7
Carbonic acid, CO_2	5·9
Nitrogen and oxygen	80·4
	100·0

The 10 cub. ft. of the combustible mixture formed 9·7 cub. ft. of burned gases of the above composition. The contraction after combustion was 3 per cent.

The London gas used by Bairstow & Alexander, of which analysis is given in the same table, required 5·3 cub. ft. of air for the complete combustion of each cub. ft. of gas.

The mixture produced by the combustion of 1 cub. ft. of London gas in 5·3 cub. ft. of air was as follows:

	Per cent. by volume
Steam, H_2O (assumed gaseous)	21·4
Carbonic acid, CO_2	8·9
Free oxygen	0·5
Nitrogen	69·2
	100·0

COMPOSITION OF EXHAUST GASES FROM COAL GAS

The exhaust gases produced by the 'X' engine of the National Gas Engine Co., Ltd., as shown by the Institution of Civil Engineers' Committee tests in 1904, were of the following composition:

	Per cent. by volume.
Steam, H_2O (assumed gaseous)	12·0
Carbonic acid, CO_2	5·0
Free oxygen	8·0
Nitrogen	75·0
	100·0

This analysis is important as illustrating the composition of exhaust gases at the coal gas mixture of nearly maximum economy, with that gas engine of 14 ins. diameter cylinder by 22 ins. stroke, when giving an indicated thermal efficiency of 35 per cent., and a brake thermal efficiency of 30 per cent.

The gas present in the charge before ignition was in the proportion of 1 vol. of coal gas to 9·2 vols. of air and exhaust gases.

In the author's experiments on the specific heat of the products of combustion the approximate composition of the gas engine exhaust tested in the same 'X' engine at Ashton-under-Lyne was:

	Per cent. by volume
Steam, H_2O (assumed gaseous)	11·9
Carbonic acid, CO_2	5·2
Free oxygen	7·9
Nitrogen	75·0
	100·0

When using mixtures with excess of oxygen present, practically no carbonic oxide is produced; but if the mixture be irregular or with excess of gas, then carbonic oxide is found, sometimes as high as 3 per cent. of the volume of exhaust gases.

The air required for complete combustion of town gas in this country varies now (1912) between the extreme limits of 5 to 6 of air for 1 of gas.

Formerly (see earlier editions of this book) the range of variation was much wider, from 5 to over 7 vols. of air for 1 of gas. In the south country the candle-power was low, about 16 candles; but in the north, where Cannel coal was accessible, it was as high as 28 candles, and the large proportion of heavy hydrocarbons increased the air required for combustion.

Producer Gas.—Producer gas may be defined as gas formed by the partial combustion of fuel in a suitable apparatus; it consists mainly of a mixture of carbonic oxide and hydrogen, with small quantities of hydrocarbons and large quantities of nitrogen together with some carbonic acid.

Ordinary town illuminating gas is produced by the destructive distillation of coal in a closed retort. The amount of gas obtained per ton of coal depends on the temperature of distillation, or the temperature of carbonisation, as the gas engineers call it. At a comparatively high temperature a larger volume of gas is given off per ton, but the illuminating power is low. A good gas coal on destructive distillation will yield at a fair carbonising temperature about 11,000 cub. ft. of gas per ton of from 15 to 17 candle-power, and it will leave in the retort from 62 to 73 per cent. of coke; that is, of 100 tons of the original gas coal, 38 to 27 tons are driven off as illuminating gas, water vapour, tar and ammonia, while 62 to 73 tons remain in the retort as coke.

Gas coal is necessarily more expensive than the fuel ordinarily used in steam boilers, and the process of distillation can only liberate from the coal such volatile matters as enter into its composition. Accordingly the heat unit of coal gas is necessarily more expensive than the heat unit of coal burned in the furnace of a steam boiler. To compete with the steam boiler and furnace in producing a gas heat unit as cheaply as a coal heat unit placed on the fire-grate, it is necessary to convert the whole of the coal into gas suitable for use in a gas engine.

At first glance it appears a difficult problem to produce inflammable gas from solid carbon, either in the form of anthracite or of coke, but the principle is simple enough. When unit weight of carbon is entirely burned in air or oxygen, carbonic acid or, more properly, carbonic anhydride, is formed, that is, the gas CO_2. This gas CO_2, if passed through a sufficient depth of incandescent carbon, is converted into carbonic oxide, which is inflammable. The chemical reaction is generally given: $CO_2 + C = 2CO$.

That is, two volumes of CO_2 combined with a sufficient weight of carbon to form carbonic oxide produce four volumes of carbonic oxide gas. For the purpose of the gas engine using a properly proportioned gas generator it may be considered that the carbon used is burned to carbonic oxide only, and not to carbonic acid. The heat evolved in the process of producing carbonic oxide from carbon is:

One pound of carbon forming CO evolves 2400 centigrade heat units, or 4320 B.Th.U.; whereas,

One pound of carbon forming CO_2 evolves 8000 centigrade heat units, or 14,400 B.Th.U.;

so that the process of the formation of carbonic oxide loses a part of the heat of the carbon, and the same weight of carbon in carbonic oxide will only produce, when the carbonic oxide is burned, 10,080 B.Th.U. instead of 14,400. Thus by passing air through incandescent carbon or coke of a sufficient depth carbonic oxide gas can be formed and the whole of the carbon transformed into an inflammable gas. The air on first coming into contact with the incandescent carbon burns a portion of it to CO_2, carbonic acid gas, and this carbonic acid on passing through a further body of incandescent carbon is reduced to the inflammable gas carbonic oxide, CO. The first stage of the process evolves all the heat of combustion, and the second stage absorbs a portion of the heat so evolved. The net result is that if the inflammable gas produced be cooled down and then burned in a gas engine cylinder, the heat evolved by the combustion will only be 70 per cent. of the heat which the solid carbon would have evolved if burned directly without preliminary conversion into gas. The 30 per cent. of heat is carried away by the carbonic oxide from the gas producer, and is lost on cooling down the gas to fit it for use in the gas engine. When air is blown through the producer the nitrogen of the air of course remains, and is mixed with the inflammable CO. This is the fundamental idea of the gas producer, and accordingly it will be found that the earlier and abortive proposals for the conversion of the entire solid fuel into gas contemplated only blowing air through a sufficient depth of carbon. Taking the composition of atmospheric air as 79 vols. nitrogen and 21 vols. oxygen (the element argon may be neglected, as it is included in the nitrogen and is very similar to it), then the best gas which could be produced in this simple manner would be that in which the whole of the oxygen was used up in forming carbonic oxide. Remembering that one vol. of oxygen gas, after combining with enough carbon to make CO, forms *two* vols. of that gas, the composition of the gas proceeding from the producer would be 79 vols. nitrogen and 42 vols. carbonic oxide, that is:

		Per cent.
79 vols. nitrogen	=	65·3
42 vols. carbonic oxide	=	34·7
		100·0

The gas obtained would consist entirely of 65·3 per cent. of nitrogen and 34·7 per cent. of carbonic oxide; this gas on combustion in the engine would evolve 70 per cent. of the heat of the original carbon. That is, if the efficiency of the producer be compared with a steam boiler, it would be equal to that of a boiler giving 70 per cent. of the heat of combustion in its furnace in the form of steam delivered at the stop valve.

Such a producer, however, would waste an entirely unnecessary amount of heat, and would give considerable practical difficulty in getting rid of the 30 per cent. of the heat of all the carbon gasified in it, the lining would be overheated, and generally the temperature of the carbon contained in the producer would become undesirably intense. A certain high temperature is required, it is true, to convert the CO_2 into CO; but if that temperature be maintained it is undesirable to go above it. Gas engineers have accordingly taken advantage of another chemical reaction to use some of this heat and produce better gas. If steam be passed over highly incandescent carbon—which carbon must, however, be kept incandescent—the oxygen of the steam unites with the carbon, and the hydrogen of the steam is liberated. The ultmate effect of the reaction is to decompose steam and produce hydrogen and carbonic oxide; the reaction is as follows:

$$H_2O + C = H_2 + CO.$$

That is, two volumes of water vapour in contact with incandescent carbon produce two volumes of hydrogen gas and two volumes of carbonic oxide gas. This reaction, however, absorbs heat to produce the decomposition of the steam; more heat requires to be absorbed than is given out by the burning of the carbon to CO.

To decompose steam containing 2 units weight of hydrogen gas requires the absorption of 68,340 centigrade heat units, or 123,010 B.Th.U., and in producing 28 units weight of CO from 12 units weight of carbon there are evolved 28,800 centigrade heat units, or 51,840 B.Th.U.; that is, the heat evolved by the carbonic oxide produced in the reaction is about one-half of the total heat required. This reaction cannot therefore proceed without a sufficient supply of heat from some source—in this case the 30 per cent. which would otherwise be lost.

The composition of the gas so produced would be:

$$\left.\begin{array}{lcl} N & = & 79 \text{ vols.} \\ CO & = & 42 \text{ ,,} \end{array}\right\} \text{ produced by the action of atmospheric air upon carbon.}$$

$$\left.\begin{array}{lcl} CO & = & 9 \text{ ,,} \\ H & = & 9 \text{ ,,} \end{array}\right\} \text{ produced by the reaction of steam upon carbon.}$$

$$139 \text{ vols.}$$

The percentage composition would be about:

	Per cent. by volume	
N =	56·8	
CO =	36·7	43·2 total combustible gases.
H =	6·5	
	100·0	

The higher heating value of this gas is about 138 B.Th.U., and the lower value 134·7 B.Th.U. per cubic foot at 16° C. and 14·7 lbs. per sq. in. pressure.

The production of a gas of this composition assumes that all the heat is utilised for the purpose of reaction and that none is lost from the apparatus. It assumes also that all the heat carried away by the gas after formation is returned to the air and steam which are about to perform the reaction. This, of course, is impossible, but the calculation has been made in order to supply a standard of comparison. Such a gas would contain the whole of the heat of the original carbon before gasifying. In an actual apparatus the carbon is placed in a brick-lined producer, ignited, and blown up to a good heat by a forced draught; the producer is then closed, and steam and air blown in, in definite proportions, to pass through the incandescent carbon mass. The resulting gas passes away from the producer in a heated state, and is cooled before being sent into the gas-holder. The reaction requires a certain temperature for its continuance, so that the interior of the producer must not fall below it; the gas is discharged at a somewhat lower temperature, but a greater supply of air is necessary than that calculated to make up for the heat losses.

Carbonic acid can be used instead of steam to absorb the heat of the reaction, resulting in the formation of carbonic oxide from the oxygen of the air.

Such a producer using carbon only gives a gas which is practically free from hydrogen except such as comes from the decomposition of the moisture in the atmosphere and from the hydrogen in the carbon fuel. The carbonic acid may be obtained from the exhaust gases of the engine, a proportion of which gases may be passed through the producer with whatever additional air may be required. In this case the gas produced would have the composition:

	Per cent. by volume
Nitrogen	65·3
Carbonic acid	34·7
	100·0

But 70 per cent. of the carbonic oxide would come from the oxygen of the atmosphere or exhaust assumed to be used to burn the carbon to CO, and 30 per cent. would come from the decomposition of the carbonic acid sent into the producer with the exhaust. That is, 24·3 per cent. would come from free oxygen and carbon and 10·4 per cent. from decomposition of carbonic acid by the spare heat of the oxidation to CO.

The exhaust from the engine would be required to mix with atmospheric air to give the following as the composition of the gaseous mixture flowing through the incandescent carbon in the producer:

Producer gas (hydrogen free) Per cent.		Mixture of gases to pass through carbon
N = 65·3	gives	N = 65·3 vols.
CO = 24·3 from O	gives $\frac{24·3}{2}$	N = 12·15 vols.
CO = 10·4 from CO_2	gives $\frac{10·4}{2}$	CO_2 = 5·2 vols.
100·0		82·65 vols.

That is, if 82·65 vols. of mixed gases containing 65·3 vols. nitrogen, 12·15 vols. oxygen, and 5·2 vols. carbonic acid be passed through incandescent carbon, there will be just enough chemical energy in the supposed absence of all losses to produce an inflammable gas containing 65·3 per cent. of nitrogen and 34·7 per cent. of carbonic oxide.

The heating value of this gas would be approximately 111 B.Th.U. per cub. ft., measured at 16° C. and 14·7 lbs. per sq. in., that is, atmospheric pressure.

The reaction $CO_2 + C = 2CO$ depends on the attainment of a certain temperature of contact between the air and carbon; at lower temperatures the reaction is reversed and the CO produces CO_2 and carbon thus:

$$2CO = CO_2 + C.$$

At 1000° C. the reaction producing CO from CO_2 is practically complete; at a temperature of about 450° C. only 2 per cent. of CO is formed, leaving 98 per cent. of CO_2.

Messrs. Dowson & Larter in their excellent book on producer gas give a table calculated from the experiments of the French chemist Boudouard by means of M. Boudouard's formula. The following numbers have been taken from that table:

PROPORTIONS OF CO_2 AND CO FORMED BY THE ACTION OF OXYGEN ON CARBON AT DIFFERENT TEMPERATURES. (*Dowson*)

Temperature Degrees C.	Per cent. by volume	
	Carbon dioxide, CO_2	Carbon monoxide, CO
446	98	2
538	90	10
623	70	30
678	50	50
731	30	70
814	10	90
861	5	95
925	2	98
977	1	99
1030	0·5	99·5

From this table it is evident that the proper working temperature of a gas producer using carbon is found between 700° and 1000°. It need not be so high as 1000° except for securing a rapid rate of conversion, but it must not be lower than 700° if excess of CO_2 is to be avoided.

The gas producers at present in use consuming anthracite and coke are of two kinds: pressure and suction. Mr. J. E. Dowson, M.Inst.C.E., was the first to succeed with a pressure producer, and he described his apparatus as applied to a 3 HP Otto gas engine at the York meeting of the British Association in 1881.

In the pressure producer steam is supplied from a small separate boiler to an air injector, by which a mixture of air and steam is forced through a mass of incandescent carbon, and the gas formed is collected in a gas-holder and from it supplied to the engine.

In the suction producer, which was first successfully applied by M. Léon Bénier of Paris in 1894, both boiler and gas-holder are dispensed with, and the engine suction is utilised to draw air and steam through the carbon in the producer, the air coming from the atmosphere and the steam produced by the waste heat evaporating water at atmospheric pressure. The air and steam flow through in proper proportion, and the gas formed is of nearly the same composition as that of the pressure apparatus.

Messrs. Dowson & Larter give the following analysis as the mean of the gas generated by seven pressure producers of Mr. Dowson's type, the gas being taken when the producer is running hot, that is, the weak gas which comes over on first starting the producer is not included.

ANALYSIS OF DOWSON PRESSURE PRODUCER GAS; MEAN OF SEVEN PLANTS. (HOT START.) (*Dowson & Larter*)

	Per cent. by volume	
Nitrogen	49·82	
Carbonic oxide	25·55	44·11 per cent. combustible gas.
Hydrogen	17·36	
Methane	1·20	
Carbonic acid	5·77	
Oxygen	0·30	
	100·00	

Higher calorific value	161 B.Th.U. per cub. ft.
Cub. ft. of gas per ton of fuel consumed in producer	180,000

Mr. M. Atkinson Adam, B.Sc., A.M.Inst.C.E., made tests of a 40 BHP suction plant of Mr. Dowson's design at Basingstoke for the author's James Forrest Lecture in 1904, and with a 'hot start' he obtained the following analysis and results :

ANALYSIS OF DOWSON SUCTION PRODUCER GAS 40 BHP PLANT. (HOT START.) (*M. Atkinson Adam*, 1904)

Fuel, anthracite

	Per cent. by volume	
Nitrogen	56·24	
Carbonic oxide	20·13	36·93 per cent. combustible gas.
Hydrogen	15·64	
Methane	1·16	
Carbonic acid	6·09	
Oxygen	0·74	
	100·00	

Higher calorific value = 135·3 B.Th.U. per cub. ft.
Lower calorific value = 127·7 B.Th.U. „ „ „
Cubic feet of gas produced per ton of fuel consumed in producer = 208,000
Cubic feet of gas produced per pound of fuel = 92·9
Thermal efficiency of producer on higher heating value of gas = 89 per cent.
Thermal efficiency of producer on lower heating value of gas = 84 per cent.
Air required for combustion of 1 cub. ft. of gas = 0·927 cub. ft.

The anthracite used was in the form of washed peas from a well-known pit in Wales ; it is known as Western Valley anthracite, and in 1904 cost 14*s.* 6*d.* per ton at the pit and about 24*s.* per ton delivered at Basingstoke. It contained 3·6 per cent. moisture and 8·76 per cent. of ash and clinker, and it evolved in the bomb calorimeter 13,890 B.Th.U. per lb.

Other samples of the same anthracite used by Mr. Adam in a similar test of a smaller plant gave 3·37 per cent. moisture, 3·975 per cent. ash and clinker, and by bomb calorimeter 14,319 B.Th.U. per pound. The steam used in the 40 BHP plant was found to be 30 lbs. per hour, and the water used for cooling the scrubbers 400 lbs. per hour or 1 gal. per rated HP hour. The rise of temperature of the scrubbers water was 40° C.

Taking this anthracite at 30*s*. per ton delivered, which is nearer to the price in 1912, 1 lb. costs 0·16*d*., and 78·3 cub. ft. are required per 10,000 B.Th.U., for which the producer uses 0·84 lb. of anthracite.

The cost of 10,000 B.Th.U. is thus 0·135*d*. nearly for heat in the form of gas (lower calorific value) from this particular Dowson suction producer.

Mr. Larter made a similar test with the 40 HP producer used by Mr. Adam, but with small gas coke instead of anthracite. The following are the leading results of six hours' test (hot start):

Analysis of Dowson Suction Producer Gas 40 BHP Plant. (Hot Start.)
(*A. T. Larter*, 1904)

Fuel, gas coke

	Per cent. by volume	
Nitrogen	55·15	
Carbonic Oxide	25·30	38·85 per cent. combustible gas.
Hydrogen	13·20	
Methane	0·35	
Carbonic Acid	5·4	
Oxygen	0·6	
	100·0	

Higher calorific value	136·2 B.Th.U. per cub. ft.
Cubic feet of gas produced per ton of fuel consumed in producer	182,700 at 0° C. and 76 mm.
Cubic feet of gas produced per lb. of fuel	81·6 at 0° C. and 760 mm.
Thermal efficiency of producer on higher heating value of gas	89 per cent.
Thermal efficiency of producer on lower heating value of gas	84 per cent.
Air required for complete combustion of 1 cub. ft. of gas	0·921 cub. ft.

The calorific value of the coke used was 12,477 B.Th.U. per lb., so that 0·935 lb. is required to produce 10,000 B.Th.U. from the generated gas. With coke at 10*s*. per ton this would cost 0·05*d*. for coke per BHP hour.

Power from coke thus costs much less than from anthracite.

If the waste heat from the pressure producer mentioned had been used to raise steam, and the separate boiler omitted, the results from the pressure producer would be the same as from the suction on thermal efficiency.

It may be taken, then, that in recent producer plants of suction or pressure type the fuel cost with anthracite is roundly 0·135*d.*, and with coke 0·05*d.*, per 10,000 B.Th.U.

This is considerably below the fuel cost of coal gas, even when sold at 1*s.* per 1000 cub. ft.

Allowing, however, for extra costs which are incurred in respect of the producer, such as wages for attendance, water for the scrubber, and producer and scrubber depreciation and repair, it may be taken that in a 40 BHP engine these costs will add about 0·1*d.*, bringing up the cost of 10,000 B.Th.U. to practically the same figure as that of coal gas at 1*s.* per 1000 cub. ft. In larger suction plants better results are obtained.

Suction plants have now been adapted to use other cheap fuels, such as charcoal, sawdust, wood chips, peat, coco-nut shells, &c.

Carbonic acid has been used in America instead of water for utilising the spare heat of the carbonic oxide reaction, but the carbonic acid system has not come into general use, either in America or elsewhere. It has distinct advantages, however, and it is capable, if well carried out, of higher efficiencies than the steam method.

Suction producers are now built up to 500 BHP in single units.

Bituminous Fuel Producers: Pressure Type.—Producers for bituminous fuel are now built of both pressure and suction types. The best-known producer of the pressure type is that developed by the late distinguished chemist, Dr. Mond.

Like all gas producers it consists essentially of a brick-lined chamber containing incandescent fuel, through which is passed a mixture of air and steam; but a larger quantity of steam is used than is usual in order to keep down the furnace temperature so as to enable the ammonia present to be recovered.

This renders it necessary to make somewhat elaborate regenerative or heat exchanging arrangements in order to diminish the heat supply required to evaporate the necessary water. The use of bituminous fuel also necessitates elaborate arrangements for scrubbing the gas to free it from tarry matters. The producer proper consists of a brick-lined inner shell, surrounded by an outer shell through which the necessary air and steam is passed to pick up waste heat from the fuel. A water lute is provided below the fire-grate into which the jacket dips; the ash and clinker fall into the water, and are readily removed while the producer is in action. The bituminous fuel is fed in from a large hopper in such manner that partial destructive distillation occurs, and the gases so formed pass through the upper part of the hot fuel before discharge so as to reduce tarry products to a minimum. Mond producers of this kind are made in units of 1000 to 2000 BHP per producer.

When it is desired to obtain the highest yield of ammonia, a large quantity of steam must be passed through the fuel bed, and then the quality of the gas issuing becomes poorer and the thermal efficiency of the producer falls. In Mr. Humphrey's paper already referred to the higher heat value of the gas flowing from the producer is estimated as 84 per cent. of the total heat of the fuel gasified.

From the other data given, the author (Clerk) was able to show in the discussion that the actual efficiency, including coal required for steam raising, was about 67 per cent.

Mr. Humphrey gave as a typical analysis of this gas:

VOLUMETRIC ANALYSES OF MOND GAS WITH AMMONIA RECOVERY.
(*Humphrey.* 1901)

Nitrogen	42·0	
Carbonic oxide	11·0	42·0 total combustible gas.
Hydrogen	29·0	
Methane	2·0	
Carbonic acid	16·0	
	100·0 volume.	

Higher calorific value = 148 B.Th.U. per cub. ft. at 15° C. and atmospheric pressure.

1 volume of the gas requires for complete combustion	1·15 vols. air
Volume of mixture before combustion at 0° C.	2·15 vols.
Volume of products cooled to 0° C. (steam assumed gaseous)	1·95 vols.
Contraction due to combustion	9·3 per cent.

The ammonia produced per ton gasified, calculated as ammonium sulphate, was 78 lbs. approximate.

Important tests were carried out by Messrs. Bone & Wheeler in 1906 upon a Mond plant with ammonia recovery, consisting of two producers and allied apparatus, which gave approximately a total gas sufficient to develop 2500 BHP.

Five trials were made, each lasting throughout a full week, and all conditions were kept constant except the relative proportions of steam to air. In the five tests the air was saturated at the respective temperatures of 60°, 65°, 70°, 75°, and 80° C. Throughout the tests the whole weight of the coal charged into the producer and that burned under the boiler was taken; the gas was sampled continuously over eight hours during each day shift; ammonia, tar vapour, and sulphur compounds in the gas were estimated daily; the tar collected in each week's test was weighed; dried ashes, clinker, and coke withdrawn from the water troughs were weighed, and soot and fine ash carried over by the gas and deposited in the mains was estimated for each trial.

Non-caking coal of a kind suitable for the producer was used throughout the trials, known as 'Collins Green washed nuts.' It was screened through a 1 in. mesh.

Analyses of effectively mixed samples showed that the moisture (as it was fed into the producer) varied between 3 to 6 per cent.

The mean calorific value of the dry coal was:

Gross value	13,880 B.Th.U. per lb. dry.
Nett value	13,340 B.Th.U. „ „ „

The following table gives the leading results of the tests made at the extreme temperatures of saturation with water vapour respectively 60° and 80° C.

TESTS OF 2500 BHP MOND GAS PRODUCERS WITH AMMONIA RECOVERY.

(*Bone & Wheeler*)

	Steam saturation temperature	60° C.	80° C.
Coal consumed in producers, cwts. per hour . .		16·51	13·21
Coal consumed in boiler, cwts. per hour . .		2·00	3·35
Coal consumed for blast steam per hour . .		—	1·35
Total carbon losses per cent.		5·8	8·4
Higher calorific value of gas, B.Th.U. per cub. ft., at 0° C. and 760 mm.		186	169·5
Lower calorific value of gas, B.Th.U. per cub. ft., at 0° C. and 760 mm.		173	153·5
Pounds of steam in blast per lb. coal gasified .		0·45	1·55
Percentage of steam decomposed . . .		87·4	40·00
Cubic feet of air, at 0° C. and 760 mm., in blast per lb. of coal gasified		36·95	37·1
Ratio of $\frac{\text{oxygen from steam}}{\text{oxygen from air}}$		0·5	0·8
Ammonia in gas as lbs. of ammonium sulphate per ton of coal		39	71·8
Thermal efficiencies	1. Including steam for blower engine	0·778	0·665
	2. Including steam for blower engine and washers .	0·715	0·604

The analyses of the gas produced at the saturation temperatures 60° and 80° C., are as follows:

MOND GAS PRODUCED AT 60° AND 80° C. SATURATION TEMPERATURES

	At 60° C.		At 80° C.	
Nitrogen	47·5		44·55	
Carbonic oxide . .	27·3	47·25 total combustible	16·05	42·2 total combustible.
Hydrogen . . .	16·6		22·65	
Methane . . .	3·35		3·50	
Carbonic acid . .	5·25		13·25	
	100·00 vols.		100·00 vols.	

	At 60° C.	At 80° C.
Higher calorific value of gas, B.Th.U. per cub. ft., at 0° C. and 760 mm.	186	169·5
Lower calorific value of gas, B.Th.U. per cub. ft., at 0° C. and 760 mm.	173	153·5
Yield of gas, cub. ft. per ton	138,250	147,500

The mean analysis of the dry fuel was:

	Per cent. by weight
Carbon	78·41
Hydrogen	5·51
Nitrogen*	1·39
Sulphur	0·83
Oxygen (by difference)	10·03
Ash	3·83
	100·00

Total volatile matter	36·2 per cent. by weight
Nett calorific value	13,340 B.Th.U. per lb.
Gross calorific value	13,880 B.Th.U. per lb.

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* The nitrogen, if wholly recovered as ammonia, would give 147 lbs. of ammonium sulphate per ton of dry coal.

From these values it will be seen that when a large quantity of steam is added to the air passing through the producer, the ammonium sulphate yield rises to 71·8 lbs. per ton gasified, but the thermal efficiency falls to 60·4 per cent., and when the smaller quantity of steam is added ammonium sulphate falls to 39 lbs. per ton, but thermal efficiency rises to 71·5 per cent. In the earlier tests of Mr. Humphrey a larger yield of sulphate was obtained with a thermal efficiency of about 67 per cent.

Taking the higher thermal efficiency of 71·5 per cent., 10,000 heat units at the lower value of 13,340 B.Th.U. per lb. of fuel require 1·125 lbs. of dry fuel, calculated on the lower calorific value of the gas produced.

This kind of non-caking bituminous slack can generally be delivered at the works at 10*s.* per ton, but it contains about 4 per cent. of moisture; allowing for this, the 10,000 B.Th.U. require 1·17 lbs. of coal as delivered. The cost of this is nearly 0·063*d*.

The sulphate of ammonia recovered amounts to 39 lbs. per ton of fuel gasified, and taking the selling price at the works at £10 per ton, this gives 0·022*d*. per 1·171 lbs., obtained for the ammonia. This should be deducted from the cost of fuel, but additional work and upkeep is required because of the sulphate plant, and we may assume that at the most this additional cost is just met by the sale of ammonia. On this assumption 10,000 B.Th.U. is given to the gas engine in the form of Mond gas at 0·063*d*.

Thus, although the thermal efficiency of the Mond producer is only 71·5 per cent. against the 89 per cent. of a Dowson suction producer as tested by Mr. M. A. Adam, it gives 10,000 heat units, at 0·063*d*. cost, against 0·135*d*. for the anthracite, or less than half cost in money.

When the Mond producer is worked at the higher saturation temperature of 80° C. the thermal efficiency falls to 60·4 per cent., and for 10,000 B.Th.U. this requires, on the same assumptions, 0·075*d.* for coal; but 71·8 lbs. of sulphate of ammonia are obtained, selling at 0·04*d.*; if it be assumed that the 0·22*d.* cost is not increased by the increased recovery this gives an advantage of 0·22 — 0·04 = 0·018*d.* from ammonia, which should be deducted from the higher fuel cost, 0·075 — 0·018 = 0·057*d.* per 10,000 heat units, so that on this basis it would pay to work at the lower thermal efficiency to get an increased yield of ammonia.

Messrs. Bone & Wheeler's trials are the most complete yet made of a bituminous plant, but other similar pressure plants are known to give like results. Thus a test made by the author (Clerk) of a 500 BHP Crossley bituminous plant in 1902 gave a coal consumption of 1·2 lbs. per BHP hour without ammonia recovery, and the coal cost was practically the same.

A Mond plant of 2000 HP is installed at the works of the National Gas Engine Co., Ltd., Ashton-under-Lyne, and it acts most successfully, giving an average consumption of about 1⅛lbs. of non-caking bituminous slack per BHP hour; this figure includes also the coal used for evaporating the steam for the producer. From each ton of coal gasified 63 lbs. of ammonia sulphate, worth £13 per ton, are obtained; as the total amount of coal gasified by the plant is 1800 tons per annum, the value of the ammonia sulphate recovered is roundly £660. A further 450 tons of coal are used per annum by the vertical boiler which supplies steam to the producer. The non-caking bituminous slack is delivered by the St. Helen's Haydock colliery at the works of the National Co. at a cost of 9*s.* 9*d.* per ton (July 1912), so that the fuel cost is only about 0·059*d.* per BHP hour.

Bituminous Fuel Producers: Suction Type.—Bituminous producers of the suction type are now entering the market, and a small number are in commercial use as made by Dowson, Crossley & Morton. In all these producers the operation is designed to destroy any tar coming from the distillation of the coal before the gas leaves and enters the coolers and scrubbers. In the pressure bituminous producer, elaborate arrangments are made for removing tar and other impurities by scrubbing, filtering, and so on, while in the suction plant the attempt is made to send gas from the producer so free from tar as to require little or no scrubbing. In the Dowson and many other plants this is accomplished by taking the gas away from the producer at the hot zone and arranging air and steam supply partly by down draught and partly by up draught. For this purpose gas is taken away by means of a hollow bridge passing through the centre of the incandescent fuel. The fuel is fed in by a hopper from above; the gases distilling from the fresh coal pass downwards with some air and steam, and the reactions which take place in the hot zone destroy tar and deliver nothing but fixed gases. The carbonised fuel passes slowly down to the grate supporting the mass and meets a current of air and steam coming through the grate and passing upwards, which current gradually produces from the carbon and steam the gases carbonic oxide and hydrogen with some carbonic acid. Gas engines actuated by gas from such producers at full power require about 1·2 lbs. of similar coal to that used in Mond plants per BHP hour, but so far no very accurate tests have been published.

A bituminous fuel suction producer of a different type has been developed by Mr. Farnham, and a prolonged test was made by the author (Clerk) from the 4th to 10th July, 1910, inclusive, which was very satisfactory. The gas engine built by the National Gas Engine Co., Ltd., developed 100 BHP continuously on a running consumption of 1·18 lbs. per BHP hour of a mixture of a highly bituminous caking coal and coke.

The Farnham producer consists of the usual brick-lined iron shell, fitted below with a mechanically operated grate which can be raised and lowered at will through a height of about 12 ins.; above the upper limit of its movement is provided a wrought iron perforated plate which can be forced through the mass of fuel right across the producer by mechanical means.

The characteristic feature of the producer is the feeding of the bituminous fuel *below* an incandescent mass of coke or coked fuel, so that the products of distillation pass upwards with air and steam, and are decomposed before reaching the top of the fuel bed. In this way the tar is entirely destroyed, and the gas passing off is so perfectly free from tar that much less scrubbing is required than is usual even in anthracite suction plants. The action is as follows: At the beginning of a week's run the producer is filled up from above with coke, which is lit up and air passed through it until it is incandescent throughout. The producer is then closed up ready to start supply for the gas engine; the perforated plate is forced across the fuel after the grate has been lifted to its upper level; the incandescent fuel is thus supported by the plate, and the grate is lowered, the fire door opened, and the bituminous fuel charged in; the door is then closed and the plate is withdrawn. The bituminous fuel is now in contact with incandescent carbon, and it begins to distil off tar and inflammable gases, which mix with the air and steam drawn through the grate by suction from the engine. In this way the engine is supplied with a gaseous mixture consisting of carbonic oxide, hydrogen, methane, and nitrogen, with some carbonic acid. Discussing the trial, the author states in his report:

'The trial was in every way satisfactory; the producer freely supplied gas for the engine, which ran day after day with monotonous regularity.

'Very briefly, the results of the trial were as follows:

'In all, the plant ran under load for forty-eight hours, and stood during six nights, about eighty-four hours, with fire alight ready for starting.

'As the result of my measurements I find that the plant consumed only 1·36 lbs. of total fuel per BHP hour, including stand-by and starting losses.

'The consumption of fuel during running I estimate at about 1·18 lbs. per BHP hour. Expressed in another way, the HP available at the brake amounts from the measurements made to 21·2 per cent. of the calorific value of the fuel fed into the producer.

'This result is noteworthy and remarkable, considering the low heating value of the highly bituminous fuel used; and indeed is comparable with results obtained in producer plants fed with anthracite coal.

'I need hardly point out that the outstanding difficulty with bituminous fuel plants is the elimination of tarry matter. Ordinarily, this is effected by a complicated and cumbrous system of scrubbers.

'Mr. Farnham has, however, succeeded in a manner which greatly impressed me in eliminating the tar with the assistance of only two small scrubbers, much smaller than are ordinarily used in suction plants consuming anthracite.

'At the end of the trial I had the gas inlet valve of the engine withdrawn, and found no tar worth mentioning. The thin film of matter adhering to the valve was scraped off, and the entire quantity (the accumulation of a week's run) could be readily placed in an ordinary envelope, and weighed less than one ounce. This slight deposit did not interfere with the working of the valve in any way. I also examined the charge inlet valve and the exhaust valve, and the piston itself, and found absolutely no trace of any tarry deposit whatever.'

The gas engine was built by the National Gas Engine Co., Ltd.; cylinder 18½ ins. diameter, and stroke 28 ins.; average revolutions during the test, 191 per minute; mean effective pressure on piston, 65 lbs. per sq. in.; average brake power, 100·4 horse.

Total BHP hours developed, 4824.

Total fuel consumed, 6573 lbs., of which 21·5 per cent. was coke.

The bituminous slack was not kept under cover, so it contained 12 per cent. moisture. Its calorific value moist was 10,710 B.Th.U. Allowing for the coke, the calorific value of the mixed fuel was 10,260 B.Th.U. per lb.

The dried bituminous fuel contained from 31·7 to 34·8 per cent. of volatile gas and tarry matter.

The cost was 5s. per ton at the pit.

The development of the bituminous fuel suction producer is of very great importance to the gas engine industry on land and of paramount importance to the introduction of marine gas engines.

With further experience all difficulties will be overcome and thermal efficiencies up to 85 per cent. will undoubtedly be attained.

Coke Oven Gas.—This gas is obtained as a by-product in the manufacture of metallurgical coke at many collieries throughout the world. According to Dr. Beilby, 16 to 18 million tons of coal were used in 1909 to produce metallurgical coke for the blast furnaces of Great Britain; of this quantity, however, only 5,870,000 tons were coked in recovery ovens from which surplus gas was available for motive power. Used in gas engines, this gas would produce 150,000 BHP for 300 days at 24 hours per day, or 300,000 BHP for 300 12-hour days per annum. The conversion to recovery ovens is increasingly rapid, and when the whole 18 million tons is so coked, 460,000 BHP at 300 24-hour days per annum could be obtained.

At present, however, but a small proportion is utilised. Coke oven gas varies considerably in composition and calorific power from hour to hour and day to day.

Mr. Robson gives examples of this from one plant in England. Three days' test of the lower calorific value showed on the first day a minimum of 370 and maximum of 410 B.Th.U. per cub. ft. at 60° F. and 30 ins. mercury; on the second day tests showed a uniform 351 B.Th.U.; and on the third day minimum 345, and maximum 366 B.Th.U.

The following table gives four analyses of this gas from different sources:

Analysis and Heating Value of Coke-oven Gas (by Volume)

—	Bargoed Colliery.	Analyses from an English plant—Mr. Robson's book.		(German) Nuremburg Co.
Nitrogen	5·00	34·79	15·68	10·5
Carbonic acid	2·01	4·10	2·88	
Oxygen	0·42	3·31	0·91	—
Carbonic oxide	5·21	5·42	5·66	7·0
Hydrogen	63·42	27·46	47·07	55·0
Methane	23·14	22·85	26·06	26·0
Heavy hydrocarbons	0·80	1·35	1·60	1·5
	100·00	99·28	99·86	100·0
Inflammable gas, per cent.	92·57	57·08	80·39	89·5
			By calorimeter	
Higher heat value per cub. ft. at 60° F. and 30 ins.	460 B.Th.U.	—	379-390 B.Th.U.	—
Lower heat value	—	—	345-366 B.Th.U.	450 B.Th.U.
Air required for complete combustion of 1 cub. ft. of gas	3·95 cub. ft.	3·15 cub. ft.	3·97 cub. ft.	4·17 cub. ft.

The lower heating value may be taken to vary from 350 to 450 B.Th.U. per cub. ft. at atmospheric temperature and pressure, and the air required for complete combustion from 3 to 4 ft. per cub. ft. of gas. This variation, together with the high percentage of hydrogen in the gas, introduces difficulties in the application of the gas to driving internal combustion engines, but happily these difficulties have been met by providing means for the ready adjustment of the gas and air inlets, and further by a new method introduced by the author to prevent pre-ignition.

An interesting installation of a modern coke oven plant is found at the Bargoed Colliery of the Powell Duffryn Steam Coal Company; it consists of 100 coke ovens of the regenerative and recovery type, in which 153,600 tons of coal is coked per annum. The total gas yield is 10,000 cub. ft. per ton, half of which is required for heating the coke ovens and the other half is available for motive power purposes. This colliery alone thus produces 940 million cub. ft. of gas per annum of the approximate composition above, which can be used in gas engines. Assuming the mean lower calorific value to be 400 B.Th.U. per cub. ft., and the heat consumption of a gas engine to be 10,000 B.Th.U. per BHP hour, this gas could develop 5228 BHP continuously throughout the 24 hours for 300 working days per annum.

Three gas engines have been installed: two by the Nuremberg Co., and one by Richardson, Westgarth & Co. The two Nuremberg engines develop between them 3600 BHP.

An interesting paper was read in 1907 before the South Wales Institute of Engineers by Mr. E. M. Hann, M.Inst.C.E., from which these particulars have been calculated.

Writing of this coke oven gas, Mr. Hann states:

'The high percentage of hydrogen makes it peculiarly liable to pre-ignition by compression, and at the time at which the installation was being considered there was great difficulty in getting any of the makers of gas engines to face the problems presented by the composition of this gas. . . .

'As a consequence of the quality of the gas the compression has to be rather low, between six and seven atmospheres, but pre-ignition is a very rare occurrence indeed. . . .

'The performance of the first gas engine set was sufficiently good to induce the company to order a second and larger set from the Nuremberg Co.'

Both engines are now successfully running, and the earlier engine has been at work since February, 1907.

Blast Furnace Gas.—In 1907 the pig iron produced in Britain was 10,114,000 tons, and in 1908, 9,057,000 tons. Assume the annual production to be ten million tons; then, if the average working year be 300 days of 24 hours each, the production of pig-iron per day is 33,333 tons. Allowing for the blast furnace gas used in the process for blowing, transport, lighting, and water supply, there remains sufficient, if used in gas engines, to produce 30 BHP per ton of pig-iron for the whole 24 hours. The total power capable of production from this source is thus 1,000,000 horse for 300 days per annum, each day of 24 hours.

If all British blast furnaces and coke ovens were arranged to permit of collection and cleaning of their surplus gas, 1,460,000 BHP could be developed continuously for 300 days per annum, each day being 24 hours. Only a very small part of this great power is at present utilised.

Blast furnace gas resembles producer gas, but it has but little hydrogen, as no steam is passed into the furnace; the heat of the reaction is required to reduce the oxide of iron to the metallic state.

The following analysis shows its approximate composition:

ANALYSIS AND HEATING VALUE OF BLAST FURNACE GAS (BY VOLUME)

—	Gas in Britain (*Robson*)		Gas in Germany (*Nuremberg Co.*)
Nitrogen	66·13	55·32	70·5
Carbonic acid . . .	6·01	6·25	
Carbonic oxide . . .	25·13	33·12	26·0
Hydrogen . . .	2·73	5·31	3·0
Methane	—	—	0·5
	100·00	100·00	100·0
Inflammable gas, per cent. .	27·86	38·43	29·5
Lower heat value per cub. ft. at 60° F. and 30 ins. .	96 B.Th.U.	132 B.Th.U.	100 B.Th.U.
Air required for complete combustion of 1 cub. ft. of gas	0·66 cub. ft.	0·92 cub. ft.	0·74 cub. ft.

Blast furnace gas, it will be observed, approximates closely to the gas which would be generated in a producer fed with air only. The small proportion of hydrogen present enables very high compression to be used without danger from pre-ignition.

Comparison of the Different Gaseous Fuels.—The calorific values and the air required for complete combustion, although they vary with the conditions, may be taken approximately as follows:

Nature of gas	Town gas	Coke oven gas	Mond gas	Pressure producer gas		Suction producer gas		Blast furnace gas
				Anthracite	Coke	Anthracite	Coke	
Lower calorific value in B.Th.U. per cub. ft. at 60° F. and 30 ins.	500	400	150	150	130	130	130	100
Air required to produce complete combustion of 1 cub.ft. of gas, cub. ft.	5·0	4·0	1·1	1·1	—	0·93	0·92	0·75
Lower calorific value of gas in 1 cub.ft. of maximum mixture at 60° F. and 30 in., in B.Th.U.	83·3	80·0	71·0	71·0	—	67·0	68·0	57·0
Ditto × 0·85	71·0	68·0	60·3	60·3	—	57·0	57·8	48·4

The table shows the varying quantities of heat which can be obtained from 1 cub. ft. of maximum explosive mixture for each gas, and it varies but little compared with the variation of heating value in the different gases. Thus although the heating value of coal gas is 500 B.Th.U., and that of blast furnace gas only 100 B.Th.U. per cub. ft., yet 1 cub. ft. of coal gas and air mixture with just enough of oxygen to burn completely at atmospheric temperature and pressure contains 83·3 B.Th.U., and 1 cub. ft. of blast furnace gas and air on the same basis contains 57 B.Th.U. The producer gas and air mixtures vary but slightly—67 to 71 B.Th.U. per cub. ft.

So much heat per cub. ft. cannot be introduced into the cylinder because of heating and some throttling of the entering charge. The maximum heat capable of being introduced per cub. ft. swept by the piston may be taken as approximately 0·85 of the values at atmospheric temperature and pressure, so that with coal gas a maximum of 75 and with blast furnace gas a maximum of 48·4 B.Th.U. could be obtained per cub. ft. of piston-swept space.

Although it is possible to introduce 75 B.Th.U. per cub. ft. as stated, it is not desirable to do so; thus in the Institution of Civil Engineers tests of the 14 ins. diameter by 22 ins. stroke National engine giving 35 per cent. indicated thermal efficiency the heat present per cub. ft. swept by the piston on its charging stroke was 53·5 B.Th.U., corresponding to 100 lbs. mean pressure per sq. in. At 30 per cent. indicated thermal efficiency the mean pressure would have been 85·7 lbs.

For mixture of maximum economy it is not desirable to introduce more than 50 B.Th.U. per cub. ft. of working stroke swept by piston in engines up to 20 ins. diameter cylinder, and in engines of 30 ins. diameter cylinder the limits lie between 35 and 40 B.Th.U. per swept cub. ft. This reduction is necessary in large cylinder engines to keep heat troubles within practicable limits. The Nuremberg Co. publish the following table giving their view of Continental practice as to heating values of the different kinds of gas and their permissible mixtures with air:

Nos.	1	2	3	4	5
Kind of gas	Blast-furnace gas	Producer gas from coke	Producer gas from anthracite	Coke oven gas	Town gas
Nett calorific value, in B.Th.U.	100	120	140	450	560
Theoretical approximate quantity of air per cub. ft. of gas, in cub. ft.	0·69	0·87	1·08	4·2	5·2
Usual excess of air, per cent.	20	20	20	40-50	40-50
Actual approximate quantity of air per cub. ft. of gas, in cub. ft.	0·83	1·04	1·3	6·3	7·8
Calorific value of 1 cub. ft. of mixture of gas and air, in B.Th.U.	55	59	61	62	64
Percentage of hydrogen per cub. ft. mixture of gas and air	1·6	4·9	6·5	7·5	5·5

The heat units given per cub. ft. of mixture are too high to express current practice if referred to volume swept by the piston, for the reasons already given.

The values for town gas and coke oven gas are higher than those usually found in British practice.

Reference has been made to the tendency to pre-ignition when hydrogen is present in large proportion, and coke oven gas is of all the most troublesome in this respect.

This difficulty has been entirely overcome by the author for the large vertical engines of the National Gas Engine Co. by introducing into the air inlet pipe at atmospheric pressure about 10 per cent. to 20 per cent. by volume of cooled exhaust gases from the exhaust chamber or pit of the engine. This expedient reduces the free oxygen and replaces it by a mixture of carbonic acid and nitrogen, which has the immediate effect of entirely suppressing all pre-ignition, even when coke oven gas is used. The inert gas addition reduces inflammability by diminishing the oxygen and by the diluting effect of the carbonic acid and nitrogen, without reducing the total mass of the charge. The application is very simple and easily made, and the maximum power of the engine is materially increased. With this device considerable over-loads are quite permissible.

The power obtainable, it will be seen, varies considerably with the particular gaseous fuel used in small cylinders up to about 20 ins. diameter, but in large cylinders the weakest gas supplies quite as much heat as the engine can deal with effectively, having regard to durability and safety.

CHAPTER VI

PETROLEUM, PETROL, AND PARAFFIN OILS, WITH A NOTE ON ALCOHOL

The gaseous fuels described in the previous chapter are mostly produced from coal in its various forms, and are, therefore, of the first importance to internal combustion engines; but liquid fuels are also of great moment in view of the recent development of the petrol engine and the increasing use of heavy and waste oils. The application of the Diesel engine to large vessels such as the 'Selandia' and Dr. Diesel's recent enthusiastic advocacy of heavy oil at the Institution of Mechanical Engineers have, however, tended to distort in the public mind the true relationship of coal and oil as fuels. It is accordingly desirable to discuss the position briefly.

According to Mr. Chiozza Money, the world's output of coal in the year 1907 was 1,117,000,000 metric tons (one thousand one hundred and seventeen million metric tons), which was produced by the different countries in the following proportions:

	Per cent.
United States of America	39
United Kingdom	24
Germany	19
All the rest of the world	18
	100

According to Mr. Joseph Shaw, K.C., chairman of the Powell Duffryn Steam Coal Co., Ltd., the world's output in 1909 was 1,098,000,000 metric tons (one thousand and ninety-eight million metric tons), and the increase in output in that year, compared with the average output of the three previous years, 1905–8, was 158,000 tons.

It may be taken broadly, therefore, that the world's coal output from 1908 to 1910 inclusive was about 1,100,000,000 metric tons (one thousand one hundred million metric tons) per annum.

According to the 1911 Report of the Californian Oilfields Co., Ltd., the world's output of mineral oil in the years 1908–10 was as follows, in barrels and metric tons:

	1908	1909	1910
Barrels . .	278,000,000	293,000,000	329,000,000
Metric tons .	37,700,000	39,740,000	44,450,000

Mr. Shaw gives the 1909 oil output as 39,812,000 metric tons. That is, the total oil outputs for the three years are respectively 3·4 per cent., 3·6 per cent., and 4·04 per cent. of the coal output, taken at the average of 1,100,000,000 metric tons of coal per annum. From this it is at once evident that by no possibility could oil displace coal for *all* purposes.

The question now arises: Could oil entirely displace coal for motive power purposes?

The total of the world's power generated from coal is not known; but a probable estimate may be made, and the steam shipping tonnage gives some clue to the total steam power in use for marine work.

According to information published by the Committee of 'Lloyd's Register of British and Foreign Shipping' the gross tonnage of the mercantile steamships of the world in 1909 was 36,473,102 (thirty-six million four hundred and seventy-three thousand one hundred and two tons), divided among 21,909 steamships, which gives an average gross tonnage 1665 per vessel. One shaft horse-power per 3·647 tons appears reasonable as an average of fast and slow vessels; this gives the shaft horse-power of the mercantile marine of the world as ten million. Assume this power to be exerted for 200 days of 24 hours each per annum, in order to allow for time of the ships in port and also for a proportion of vessels laid up for a portion of the year; then 10,000,000 shaft HP would be exerted for 4800 hours per annum. 10,000,000 × 4800 = 48,000,000,000 shaft HP hours.

The warships of the world are capable of developing a much larger maximum power than the whole of the merchant navies; thus the maximum available horse-power of the war vessels of Britain, France, Germany, and the United States of America is roughly nine millions.

Of this the British Royal Navy accounts for 4·75 millions. The total war navies of the world can develop about 13 millions of horse-power. In times of peace, however, the average power exerted will be about 3 million horse. Assuming this power to be required for 4800 hours per annum, 3,000,000 × 4800 = 14,400,000,000 shaft HP hours per annum; and 14,400,000,000 + 48,000,000,000 = 62,400,000,000 total shaft HP hours required per annum.

On this estimate the total for marine purposes is 62,400 millions of shaft HP hours per annum.

The coal required for this power, at 2 lbs. per shaft HP per hour, would be 56·5 million metric tons.

The oil required, if consumed in Diesel engines, would be about 0·5 lb. per shaft horse-power hour, or $\frac{56{\cdot}5}{4} = 14{\cdot}12$ million metric tons.

From the evidence given by Dr. G. T. Beilby, F.R.S., before the Royal Commission on Coal Supplies it appears that in 1903 the railways of the United Kingdom used for all purposes 13 million tons of coal, which amounts to 8·2 per cent. of the whole domestic consumption. An earlier Royal Commission found that in the year 1869 the railways consumed 2,027,500 tons, or 2·6 per cent. of the total home consumption.

The coal consumption on railways has obviously greatly increased in the elapsed thirty-four years, and no doubt continued to increase between 1903 and 1910. If eight per cent. be taken as the proportion for 1910, it is probably under the real value.

It seems probable that the proportion of coal consumed for railways in other countries is not less than in Britain, indeed in countries such as the United States and Canada, where the railway mileage is very large, the proportion may be greater.

It appears reasonable to assume that 8 per cent. of the world's coal output is consumed in railway work for all railway purposes. It would thus amount to 88 million tons.

To calculate the equivalent in oil it is necessary to remember that locomotives consume about 2½ lbs. of coal per HP hour, and that some coal is used by the railways for other purposes than locomotion. The oil consumption of Diesel locomotives would probably be one-sixth of the coal for the power developed.

This gives 14⅔ million tons of oil per annum to generate power for the world's locomotives.

It now remains to consider the motive power required for the world's factories. Dr. Beilby estimates that 45 million tons of coal were consumed to produce the motive power required for the factories of the United Kingdom in 1903. Assuming that no increase has occurred in 1910, then 16·8 per cent. of the coal output of Britain is consumed for factory engines.

If factory engines throughout the world consume the same proportion, then their total annual consumption is nearly 185 million tons. Assuming the rate for the world to be half that of this country, there is still the large number of 92·5 million tons. This is obviously too low, as when the British consumption of 45 million tons is deducted it only leaves 47·5 millions for the factories of the rest of the world.

Taking the oil required as one-fourth, we get 23 million tons as the minimum to be obtained by using Diesel oil engines instead of steam, The position then is shown in the table following:

ALTERNATIVE CONSUMPTION OF COAL OR OIL FOR THE WORLD'S MOTIVE POWER PER ANNUM

	Coal	Oil
Marine engines .	56·5 million metric tons	14·12 million metric tons
Railway engines .	88·0 " " "	14·33 " " "
Factory engines .	92·5 " " "	23·00 " " "
	237·0 " " "	51·45 " " "

This number of 237,000,000 metric tons of coal per annum for the total world's motive power (excluding motor car engines) is probably considerably under the true consumption, but even this lower limit of value requires an oil equivalent of about 51·5 million metric tons, and the total oil output of the world is only 44·5 million metric tons. But only a small part of this oil is available for power; out of this total the demand for lamp oil, lubricating oil, and petrol for motor cars must be met, leaving about 11 million tons only of heavy oil of a price and quality suitable for Diesel engines. But even this amount of use would raise the price of fuel oil far above 50*s.* per ton (its British price in 1912),[1] so that the use of oil for more than 20 per cent. of the world's power appears improbable. There is without doubt a large and valuable field for heavy oil engines, but the recent expectations of the public as to the replacement of coal by oil are, to say the least, much too sanguine.

The world must mainly depend on coal for its motive power, whether it be used in steam or gas engines.

Light and Heavy Oil Engines.—Oil engines resemble gas engines in this, that the power is generated by the explosion of a compressed inflammable gaseous or vapour mixture in an engine operating in accordance with the four- or the two-stroke cycle.

The use of light oil, now known as petrol, is more easy than heavy oils, because of its greater volatility, and accordingly the light oil engines earlier attained a high measure of perfection. Both types, however, are now well understood; but as a knowledge of the properties of the principal hydrocarbons used will assist the engineer in deciding between differing methods of procedure, the properties of the various hydrocarbons shall now be shortly discussed from the point of view of the oil engine inventor or designer.

Chemistry of Petroleum and Paraffin Oils.—The petroleum oils used for explosion-engine purposes come mainly from the United States of America, Russia, Borneo, Java and Sumatra, Galicia, Roumania,

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[1] According to Mr. T. B. Broune the price has already been raised to about 80*s.* per ton (Pres. Address, Inst. A.E., Oct. 1912).

India, and Mexico. Paraffin oil is produced in Scotland by the distillation of shale.

Crude petroleum, formed in the course of ages from extensive animal and vegetable deposits, is the parent of the many forms of gaseous, liquid, and solid 'bituminous' substance found in nature. It varies considerably in appearance and constitution with the district from which it is obtained, being sometimes pale in colour and very mobile, at others nearly black and very viscid. On first issuing from the wells it is a mixture of many different compounds and holds various gases and solids in solution; its specific gravity ranges from 0·77 to 1·06. The various gases, liquids, and solids entering into its composition have this, however, in common: they are all hydrocarbons, that is chemical compounds of which hydrogen and carbon are the sole constituents.

Crude petroleum is in general separated by various processes of distillation, &c., into the following seven main groups of commercial products:—

1. Petroleum ethers.
2. Naphthas, benzines,[1] petroleum spirits, or 'petrols.'
3. Illuminating oils, burning oils, or kerosenes.
4. Intermediate, or gas oils.
5. Lubricating oils.
6. Residuum (America) or Ostatki (Russia).
7. Paraffin wax—where present in the crude oil.

American petroleum consists principally of hydrocarbons belonging to a chemical series known as the paraffin series. This series has the general formula C_nH_{2n+2}. Members of another chemical series, however, are mixed with the paraffin group. This other series is known as the olefine series, of which the general formula is C_nH_{2n}.

Both the paraffin and the olefine series comprise substances ranging from the gaseous state to the solid state; that is, each series contains substances which are solid, substances which are liquid, and substances which are gaseous.

The lightest member of the paraffin series is the well-known marsh gas or methane (CH_4), and one of the heaviest of the liquid products is known as pentadecane, $C_{15}H_{32}$, and the solid paraffin wax so well known in commerce in the form of paraffin candles is a mixture consisting principally of solid members of the paraffin series, together with some solid members of the olefine series. The olefine series likewise comprises a whole range of compounds beginning with the well-known gas ethylene (olefiant gas), C_2H_4, and terminating with solid olefines containing more than 20 equivalents of carbon to 40 equivalents of hydrogen.

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[1] Benz*i*ne is a commercial name of petroleum spirit.

Benz*e*ne is the chemical name of a family of compounds obtained commercially as by-products in coal-gas manufacture (*see* 'Benzol,' p. 450).

Crude Pennsylvania petroleum as it issues from the wells gives off as gases:

Methane (marsh gas)	CH_4
Ethane	C_2H_6
Propane	C_3H_8

and at least twelve separate hydrocarbons of the paraffin series have been isolated from the crude liquid. These twelve hydrocarbons are given in the table below, with formula, specific gravity, and boiling-point of each.

All these hydrocarbons, except the first, are liquid at ordinary temperatures. The boiling-points of the hydrocarbons vary from 0° C. to 260° C., and the specific gravity from 0·65 to 0·792.

It will be observed that in every one of these compounds the hydrogen atoms going to form the molecule are double the number of carbon atoms, plus an additional two hydrogen atoms. Marsh gas, for example, has in the molecule 1 atom carbon, and 2 atoms hydrogen + 2. Ethane has 2 atoms carbon, and 4 atoms hydrogen + 2, that is 6. The same proportion is given in all the members of the series in the table.

SOME HYDROCARBONS OF THE PARAFFIN SERIES FOUND IN PENNSYLVANIA PETROLEUM. (*Redwood*)

Name	Formula	Specific gravity	Boiling-point	
		Normal.	Normal	Iso.
Butane	C_4H_{10}	0·645 at 0° C.	0° C.	—
Pentane	C_5H_{12}	0·645 „ 0° C.	38° C.	30° C.
Hexane	C_6H_{14}	0·63 „ 17° C.	69° C.	61° C.
Heptane	C_7H_{16}	0·712 „ 16° C.	98° C.	91° C.
Octane	C_8H_{18}	0·726	124° C.	118° C.
			Boiling point	
Nonane	C_9H_{20}	0·71 at 15° C.	136° to 138° C.	
Decane	$C_{10}H_{22}$	0·757 „ 15° C.	160° „ 162° C.	
Endecane	$C_{11}H_{24}$	0·765 „ 16° C.	180° „ 184° C.	
Dodecane	$C_{12}H_{26}$	0·766 „ 20° C.	196° „ 200° C.	
Tridecane	$C_{13}H_{28}$	0·792 „ 20° C.	216° „ 218° C.	
Tetradecane	$C_{14}H_{30}$		236° „ 240° C.	
Pentadecane	$C_{15}H_{32}$		250° „ 260° C.	

Pentadecane, the highest here shown, has 15 atoms carbon associated with 30 + 2 atoms of hydrogen.

The hydrocarbons of this series resemble each other very much in chemical and physical properties. They decompose under the action of heat in a similar manner, and they have similar physical properties. Chemists call such a series of compounds a *homologous* series.

The American refined lamp oils of commerce consist principally of the heavier hydrocarbons given in the list, but they also contain in smaller quantity hydrocarbons of the olefine series.

The table below gives a few of the best-known members of the olefine series.

These compounds form what chemists call an *isomeric* series, because, as will be observed, they are all of the same percentage composition. Each hydrocarbon of the series contains exactly the same proportion of hydrogen and carbon, namely, 85·7 carbon to 14·3 hydrogen. The compounds, however, differ in molecular density, and this is found by the increasing vapour density; thus, if one volume of ethylene be taken as the unit of weight, an equal volume of butylene weighs 2, hexylene 3, and so on.

Some Members of the Olefine Series

		Boiling-point	Specific gravity
Ethylene (olefiant gas)	C_2H_4	Gaseous	
Propylene	C_3H_6	,,	
Butylene	C_4H_8	4° C.	
Amylene	C_5H_{10}	73° C.	
Hexylene	C_6H_{12}	70° C.	
Heptylene	C_7H_{14}	84° C.	0·714 at 0° C.
Octylene	C_8H_{16}	119° C.	
Diamylene	$C_{10}H_{20}$	165° C.	0·777 at 0° C.
Triamylene	$C_{15}H_{30}$	248° C.	
Tetramylene	$C_{20}H_{40}$	above 390° C.	

The term *isomer* is sometimes limited to compounds of the same molecular density as well as the same percentage composition. Such compounds, however, differ in physical and chemical properties.

At first it is very surprising to find that two chemical substances of identical chemical composition and molecular density, that is, with the exact proportions of the element present, the same in both, should have different properties, but the case is strictly analogous to what is known of the elements. Many chemical elements are known to exist in several forms, without change of chemical composition. Carbon exists, for example, in three forms, the diamond, graphite, and charcoal. These three forms are widely different in appearance and physical properties, but each contains nothing but carbon, and produces nothing but carbonic acid on burning.

Phosphorus also exists in two forms, yellow and red, and it is more than suspected that iron exists in several forms.

When elements vary in this way, the variations from the best-known form are called *allotropes* or *allotropic modifications*. When a chemical compound has several varieties, the variations are known as *isomers*. The word *isomer*, however, is more strictly used to denote compounds not only of the same percentage composition, but of the same molecular weight.

Bodies of the same percentage composition and different molecular weights are known as *polymers*. The olefine series then are *polymers*.

The olefines are present in American petroleum to only a small extent, but in Russian petroleum they usually occur in a rather larger proportion. The hydrocarbons present in Russian petroleum are not quite the same as the normal olefines, but appear to be isomeric modifications of the true olefine series, having the general form of $C_nH_{2n-6}H_6$. This formula seems to be a roundabout way of expressing the same thing as C_nH_{2n}, because 6H is deducted, and 6H added. It is not, however, the same form as C_nH_{2n}, but expresses chemical relationship to another set of compounds. The compounds of the general form $C_nH_{2n-6}H_6$ are called naphthenes, and the naphthenes, although of the same percentage composition as the olefines, resemble the paraffins more closely in their chemical decompositions. The naphthenes which have been isolated from Russian petroleums are, according to Redmond, as follows:

NAPHTHENES ISOLATED FROM RUSSIAN PETROLEUM

C_8H_{16}	119° C.	$C_{12}H_{24}$	196° C.
C_9H_{18}	136° C.	$C_{14}H_{28}$	240° C.
$C_{10}H_{20}$	161° C.	$C_{15}H_{30}$	247° C.
$C_{11}H_{22}$	180° C.		

The specific gravity of the first-mentioned hydrocarbon octonaphthene, C_8H_{16}, at 0° C. is 0·7714, and that of dodecanaphthene at 17° C. is 0·8027.

Paraffin oil, as its name implies, is mostly composed of members of the paraffin series, and it is produced by the destructive distillation of Scottish shale. The crude oil obtained from the retorts contains, like crude petroleum, substances both solid, liquid, and gaseous. The solid paraffin of commerce is largely obtained from this paraffin oil.

The chemistry of petroleum and paraffin oils is extremely complex, and only a general idea has been here given of the main features.

Before leaving the chemistry, it is desirable to consider the decomposition of these compounds by heat. It is found, for example, that if a heavy member of the paraffin series be exposed to heat under pressure, so as to attain a temperature higher than its normal boiling-point, then that compound decomposes into a lower paraffin and an olefine. The paraffin hydrocarbon $C_{12}H_{26}$, for example, may be decomposed into hexylene C_6H_{12}, and hexane C_6H_{14}. The reaction may be taken as follows:

$$C_{12}H_{26} = C_6H_{12} + C_6H_{14}.$$

The heavier hydrocarbon thus splits up into an olefine and a paraffin containing a smaller number of carbon and hydrogen equivalents to the molecule. It depends entirely, however, on the particular temperature and treatment as to the actual decomposition which will take place. If the temperature of the hydrocarbon be raised to a high enough point, marsh gas, CH_4, can be produced, and carbon left in the retort. The olefines decompose also, heavier olefines producing lighter olefines by the influence of heat, or lighter olefines together with hydrogen, marsh gas, and solid carbon deposit.

Petroleum Ether and Spirit.—The volatile liquids produced from American petroleum have been classed as petroleum ether and petroleum spirit. The following table gives a list of the substances so produced. The names given are not chemical names, but ordinary trade names, and the compounds are not pure hydrocarbons of one composition, but mixtures of hydrocarbons boiling at very low points.

PETROLEUM ETHERS AND SPIRITS

Class	Commercial name	Specific gravity at 32° F.
Petroleum ethers.	1. Cymogene	0·590
	2. Rhizoline	0·625 to 0·631
	3. Gasolene	0·635 „ 0·666
Petroleum spirits.	4. C Naphtha (benzine naphtha) .	0·678 „ 0·700
	5. B Naphtha	0·714 „ 0·718
	6. A Naphtha (benzine) . . .	0·741 „ 0·745

According to Mr. Alfred H. Allen, cymogene consists chiefly of butane, C_4H_{10}, of pentane, C_5H_{12}, and an isomer of that substance; and hexylene, C_6H_{12}, and an isomer of hexylene.

All these products are extremely volatile, cymogene boiling at 32° F., the freezing-point of water, and the heaviest A naphtha boiling away under 160° F. It follows that they are dangerous to handle, and require care when used in oil engines. The substance cymogene, for example, could only be retained in the liquid state permanently by means of a freezing mixture, and all the others are so volatile that it would be dangerous to approach an open vessel containing them with a light. Any one of these liquids would take fire instantaneously on plunging a lighted match or taper into the liquid.

They are all clear limpid fluids, having when pure a rather agreeable odour.

Boiling-point of Crude Petroleum.—Crude oil being thus a mixture of a large number of hydrocarbons of very different degrees of volatility, no constant boiling-point is definable; some of the lightest (i.e. gaseous) constituents, in fact, evaporate away rapidly from the crude oil on exposure to the air at ordinary temperatures.

The proportions of distillate obtained at various definite temperatures also vary very considerably in different samples, some typical results being given in the following short table.

Proportions of Distillate from Crude Petroleum at Various Temperatures. (*From Engler and Levin*)

Country of origin	Specific gravity at 17° C. (63° F.)	Distillation commenced at ° C.	Volume percentages distilling at temperature Cent.										
			Below 130° C. (266° F.)	130° to 150°	150° to 170°	170° to 190°	190° to 210°	210° to 230°	230° to 250°	250° to 270°	270° to 290°	290° to 300°	Volume remaining above 300° C.
Pennsylvania	0·8175	82°	15	6	5	5	5	5·8	4·8	6	4·8	2	40·6
„	0·8010	74°	24·5	7	4·5	4·5	6·5	5	4·8	3·3	4	2·5	33·3
Galicia . .	0.8235	90°	16	10·5	10·3	6·5	6·5	7	6·8	6	3·5	0·5	26·4
Baku . . .	0·8590	91°	16	7	6·5	6·5	5	5	5	5·5	3·5	1	39
„ . . .	0·8710	105°	3·8	4·8	5·5	4·8	5·3	5	7	4·8	5·5	1·8	51·7
Elsass . .	0·9075	135°	—	3	4·4	5·4	4·5	6·6	7·3	7	10·3	4·5	47
Hanover . .	0·8990	170°	—	—	—	4·8	5·3	6	4	5	5	2	67·9

Determination of Specific Gravity.—Commercially, especially with American oils, the specific gravity of petroleum and its liquid products is commonly expressed in degrees of the Beaumé hydrometer. Denoting by B the Beaumé figure, the relation between specific gravity and B is expressed by the formula:

$$\text{Sp. Gr.} = \frac{140}{130 + \text{B}} \qquad (1)$$

Thus pure water is 10° B, while 0·636 gasolene is 90° B.

'PETROL'

The present importance of the formerly almost waste substances included under this term necessitates a more detailed account of them being given.

The word 'Petrol' began to come into general use about 1898, earlier alternative names being 'Light oil,' 'Mineral Spirit,' and 'Motor Car Spirit' in Great Britain; 'Motor Naphtha,' 'Motor essence,' 'Essence,' 'Oleonaphtha,' 'Automobiline' and 'Stelline' on the Continent; while in America the word 'Gasolene' was early applied, and has been retained.

Rather more than one-half of the petrol used in Great Britain in 1909 came from the Dutch East Indies per the Asiatic, Royal Dutch, and British Petroleum Companies respectively; the United States came next in order and supplied about one-fourth of the whole; a largely increased quantity from this source is anticipated in the near future. Russia has, so far, supplied us with but little, and here again a large increase may be expected from 1911 onwards; the Grozni Petroleum Company has recently made arrangements, for example, to supply the British market with about six and a quarter million gallons annually. Considerable supplies may also be expected before long of Burmese benzine, while South America is also preparing to enter the field as a supplier.

In the table on p. 446 the official figures of imports for the years 1905–1910 inclusive are given.

Some of this petrol is used by dyers and cleaners and india-rubber manufacturers, but the great bulk is burned in the engines of automobiles; the very rapid increase in the quantity used well illustrates the marvellous development of the automobile industry; it must be remembered also that the petrol consumption elsewhere, notably in the near Continent and in the United States, is very large; the diagram on p. 447 (fig. 275) illustrates the growth of the import during the six years 1905 to 1910, both inclusive.

IMPORTS OF PETROL INTO GREAT BRITAIN

Country of origin	Gallons imported during year					
	1905	1906	1907	1908	1909	1910
Dutch E. Indies . .	8,113,273	16,444,283	22,652,560	24,266,094	29,075,016	25,537,818
United States . .	10,527,066	7,988,229	7,171,850	6,097,096	15,096,918	20,721,450
Roumania . . .	—	2,303,696	1,459,000	6,822,307	4,741,970	3,644,125
Russia . . .	110	7,930	321,690	4,048,790	1,710,084	1,279,818
Germany . . .	—	32,753	176	672	1,800	—
Mexico . . .	—	—	—	—	409,623	735,908
Netherlands . . .	—	—	—	—	—	3,129,973
Other countries . .	17,942	15,796	38,329	573,036	887,870	118
Totals . . .	18,658,391	26,792,687	31,643,605	41,807,995	51,923,281	55,049,210

The great inflammability of petrol and petrol vapour has rendered necessary the framing of special regulations controlling its storage and use; unlike alcohol it is not miscible with water, which is accordingly useless in quenching a petrol fire; sand, earth, and non-combustible absorbent materials generally are best for the purpose.

The Regulations, dated July 31, 1907, of the Secretary of State, under Section 5 of the Locomotives on Highways Act, 1896, require, *inter alia*: [1]

1. That the amount of petrol stored shall not exceed 60 gallons, including that contained in the tanks of the car, in any one storehouse.

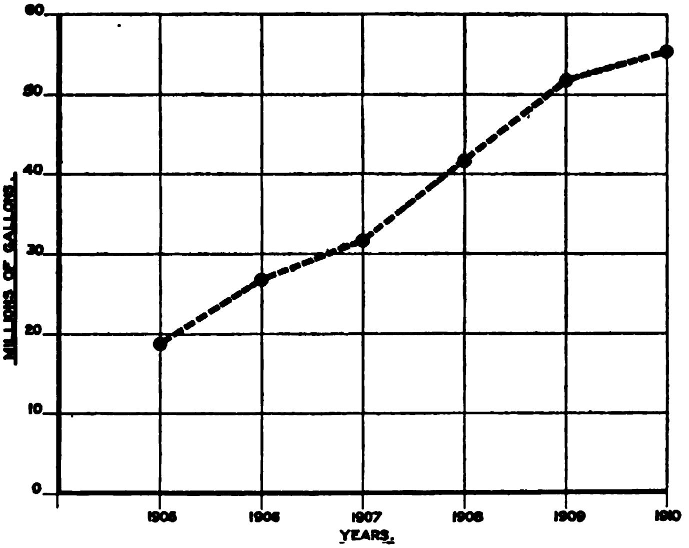

FIG. 275

2. In the event of a storehouse being within 20 feet of any other building, whether in the occupation or not of the person storing the petrol, or within 20 feet of any timber stack or other inflammable goods, notice shall be given to the local authorities under the Petroleum Act, 1871. This restriction does not apply to petrol kept in the tank of the car. Every storehouse shall be thoroughly ventilated.

3. Two storehouses within 20 feet of each other are deemed to be one, and therefore only 60 gallons may be stored in buildings so placed.

4. The storehouse shall not form part of a dwelling or be in connection with a place where persons assemble.

5. In a storehouse or in any place where a light locomotive is kept,

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[1] From Critchley's *British Motor Vehicles*, 1911 (Clayton).

petroleum spirit shall not be used for cleaning or lighting or for any purpose other than as fuel for the engine.

6. All vessels not forming part of a car when used for keeping or conveying petrol shall bear the words 'Petroleum Spirit, highly inflammable.'

7. Petroleum shall not be allowed to escape into any inlet or drain communicating with a sewer.

In the early days of the automobile industry it was considered necessary to use petrol having a specific gravity of about 0·68, but with progress in refining and improvements in carburetting the gravity was gradually raised to 0·71 or 0·72 for the engines of ordinary touring cars, and a much larger quantity then became available for use; in the engines of motor omnibuses and commercial vehicles Borneo benzine, or 'Crown' spirit of specific gravity 0·76 is now largely used; this is about the heaviest spirit at present generally employed. A benzine of 0·78 gravity is imported by the British Petroleum Co., but is at present used by cleaners and certain manufacturers only.

Among the earliest systematic experiments upon commercial petrol are those of Mr. G. H. Baillie, whose results are set forth in a paper presented on May 14, 1908, to the Royal Automobile Club. Mr. Baillie thus states the qualities desirable in a fuel for automobile engines:

1. Easy starting from cold.
2. Absence of objectionable odour, both in the fuel itself and in the exhaust products.
3. No liability to form deposits in cylinders or valve passages.
4. Economical consumption.

Ideal petrol would be of homogeneous composition with a constant boiling-point and unvarying specific gravity at that boiling-point; actual petrol departs very far from this ideal perfection, as is clearly illustrated in fig. 276, drawn from some of Mr. Baillie's results, in which abscissæ represent percentages of the original sample distilled off at temperatures given by the corresponding ordinates.

The definite members of the paraffin series of hydrocarbons known as hexane (C_6H_{14}) and heptane (C_7H_{16}) are largely contained in ordinary petrols, and octane (C_8H_{18}) is also usually present; these have constant boiling-points of 68·5° C., 98° C., and 120° C.; the corresponding curves accordingly appear as horizontal straight lines in fig. 276, and are introduced to show the contrast between the behaviour under fractional distillation of the samples of 'petrol' tested, with that of hydrocarbons of definite composition.

The 0·77 benzol furnishes a very fairly horizontal line between 80° C. and 85° C., and in this respect differs notably from all the other fuels tested excepting that marked '0·69 A petrol,' which is almost parallel to it, suggesting some similarity in constitution. This A petrol was the fuel adopted in the 1907 Circuit des Ardennes Competition, and in reference to this and the fuel marked 'B petrol' in fig. 276 Mr. Baillie said:

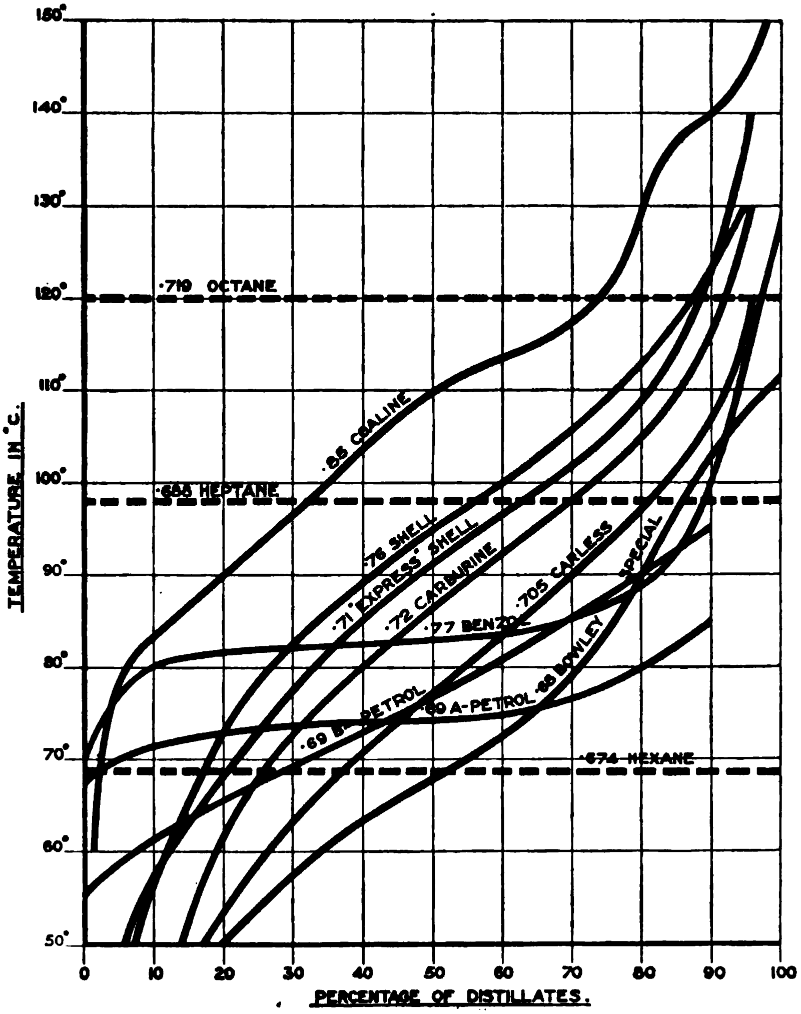

FIG. 276

'Petrol A is remarkably good, the specific gravity of the last tenth being only 0·715 and its BP only 85°. The specific gravity of the whole petrol was 0·690. Petrol B had the same specific gravity, and so by the specific gravity test would be reckoned equally good, but the curves show that its composition is very different. The last tenth has a specific gravity of 0·747, and the mean specific gravity has been brought low by adding or retaining some very light spirit of 0·650 specific gravity. . . . Petrol B is a mixture of heavy and light spirit, and, though a good spirit from the English point of view, was regarded as bad in the French book from which I took the curves. The very light spirit which is retained to get round the specific gravity test is often useless, and may be harmful. If the carburettor is water-jacketed, or becomes warm from being near the engine, the light spirit vaporises in the float-chamber and is lost, or vaporises in the passage to the jet, and then the bubbles of gas are liable to interfere with the flow of petrol in the jet.'

The 'coaline' referred to in fig. 276 is a 'benzol' spirit produced by the Coalite Company; the 'benzol' was supplied by the South Metropolitan Gas Company.

Ordinary 'benzol' consists principally of a mixture of the benzenes (C_nH_{2n-6}) known as normal benzene (C_6H_6), toluene (C_7H_8), and occasionally xylene (C_8H_{10}).

Each of these substances has a constant boiling-point and can be obtained in a pure condition, e.g. pure benzene is employed in the preparation of aniline dyes. Commercial benzol is thus a much more nearly homogeneous substance than ordinary petrol, as is clearly indicated by comparison of the curves in fig. 276. Ordinary benzol is a neutral, colourless, limpid liquid having a slightly 'burnt' odour; it is a heavy spirit, its density at 60° F. being sometimes as much as 0·88; it is very inflammable, its flash point being below 32° F. Its ascensional capillary power at 60° F. is about three and a half times as great as that of ordinary petrol; it solidifies at about 20° F. into a greasy mass; its calorific value (lower) is about 17,250 B.Th.U. per lb.

The very volatile substance hexane, present in most petrols, greatly facilitates starting from cold; butane (C_4H_{10}), which boils just below the freezing-point of water, and is accordinglygaseous at ordinary temperatures, frequently exists as a dissolved gas in small quantities in petrol, and also assists in starting; in the early surface carburettors the difficulty of starting on 'stale' petrol arose from these light constituents of the petrol having been lost by evaporation.

Surface carburettors supplied a true gaseous mixture to the cylinders, whence the importance with this type of using a light, easily volatilised spirit as, e.g. 'Carless' or 'Bowley'; with spray carburettors the whole of the petrol is, by the suction, mingled with the inrushing air, and no selective action is possible; the mixture reaching the cylinders may be partly gaseous and partly suspended spray, but such mixtures are easily explosive, though it is, of course, always desirable that the mixture entering the engine should be entirely in the form of gas.

The latent heat of evaporation of the petrol requires that a supply of heat shall be available, and accordingly most modern carburettors are hot-jacketed, or special arrangements are made to effectually warm the air before admission to the carburettor. Mr. I. I. Redwood ('Mineral Oil,' 1897) gives the following particulars regarding the latent heat of some light petroleum spirits:

LATENT HEAT OF PETROLEUM SPIRITS. (*I. I. Redwood*)

Substance	Specific gravity	Boiling-point	Latent heat B.Th.U. per lb.	Vapour density	Specific heat
Gasolene . .	0·642	70° F.	100·2	—	0·5800
Naphtha . .	0·720	115° F.	100·6	3·005	0·5687
Naphtha . .	0·756	175° F.	103·5	3·190	0·5104
Burning oil . .	0·810	260° F.	105·4	4·270	0·4991

For example, to raise 1 lb. of 0·72 naphtha from 50° F. to 115° F., and convert it into vapour at 115° F., will require, according to these figures:

$$(115 - 50) \times 0{\cdot}5687 + 1 \times 100{\cdot}6 = 137{\cdot}5 \text{ B.Th.U.}$$

Several series of careful experiments to determine the physical properties of some commercial petrols were made by Mr. J. S. G. Thomas and Dr. W. Watson, F.R.S., and their results are given in the *Proc. Inst. A.E.*, Vol. III, pp. 429 *et seq.*; the samples tested, excepting the Pratt, Carless, Carburine, and Shell, were the same as those used by Mr. Baillie already referred to. The calorific value was determined by burning the vapour in a Bunsen burner within a Boys' calorimeter; for purposes of comparison figures are also given for the definite paraffin hydrocarbons pentane (C_5H_{12}), hexane, and heptane; some of the results are exhibited in the table on p. 452.

Over the range from 5° C. to 25° C. (say, 40° F. to 80° F.), the relation between specific gravity and temperature is, for all practical purposes, expressed by the simple formula:

$$D = d\,\{1 - a\,(T - 15)\} \qquad (2)$$

where D is the required specific gravity at T° C., d is the specific gravity at 15° C., and a is the mean coefficient of expansion as given in the third column of the table.

The vapour density at 0° C. and 760 mm. pressure was deduced by calculation on the assumption that the vapour acts as a perfect gas and could be cooled to this temperature without liquefaction.

PHYSICAL PROPERTIES OF SOME COMMERCIAL PETROLS. (*From Thomas and Watson's Experiments*)

Description of petrol	Specific gravity at 15° C. (59° F.)	Mean coefficient of expansion at 15° C.	Vapour density at 0° C. and 760 mm. pressure air = 1	Calorific value (lower)	
				B.Th.U. per lb.	B.Th.U. per gallon at 15° C.
Pentane	0·630	0·00155	2·51	18,410	116,300
Hexane	0·680	0·00133	2·99	18,770	127,900
Heptane	0·699	0·00100	3·45	18,702	138,100
Bowley's special	0·684	0·00131	3·05	19,190	131,500
Carless	0·704	0·00131	3·11	18,760	132,300
Express	0·707	0·00100	3·35	18,040	127,600
Ross	0·714	0·00133	3·33	18,670	133,600
Pratt (*a*)	0·719	0·00125	3·16	18,610	134,100
Pratt (*b*)	0·720	0·00121	3·20	18,590	134,200
Carburine	0·720	0·00121	3·28	18,680	135,000
Shell (ordinary)	0·721	0·00121	3·27	18,720	135,300
Dynol	0·725	0·00145	3·43	18,520	134,600
Simcar benzol	0·762	0·00111	3·24	17,080	130,400
0·760 (Baillie)	0·767	0·00105	3·29	18,540	142,500
0·760 Shell	0·767	0·00105	3·36	18,250	140,300
Coaline	0·846	0·00109	3·31	16,690	141,500

When the differences in the origin, mode of preparation, and chemical constitution of the various petrols are borne in mind, the small variation in the calorific value per lb. is interesting; for the more usual qualities the value departs but little from 18,600 B.Th.U.

It may be mentioned here that 'Dynol' and 'Ross' are spirits obtained from the distillation of Scotch shale.

Some valuable experiments on the properties of ordinary petrols have also been made by Mr. B. Blount, whose results are given in Volume III of the *Proc. Inst. A.E.*, pp. 301 *et seq.*; a selection from these is given in the accompanying table.

The calorific values were obtained with a bomb calorimeter, the sample of petrol under test being placed in a rather deep cup provided with a celluloid cover rising above the edge and contracted at the top into a somewhat narrow mouth; in this way the vapour was caused to burn comparatively slowly, and imperfect combustion and violent explosions were avoided; a correction was made for the heat of combustion of the celluloid cover. It will be observed that all Mr. Blount's calorific values obtained with this apparatus are somewhat higher than those found by Messrs. Thomas and Watson, and also that the value appears as nearly constant throughout, and at an average of about 20,200 B.Th.U. per lb.

FRACTIONAL DISTILLATION AND CALORIFIC VALUE OF SOME PETROLS
(*From Mr. B. Blount's Experiments*)

Description of petrol	'Anglo' 760		'Shell'		Pratt's motor spirit		Carless' 'Standard'		Carburine		Russian petrol	
Specific gravity of sample .		0·739		0·717		0·715		0·700		0·717		0·705
Distillation :												
Commenced at . . .	70° C.		65° C.		63° C.		56° C.		65° C.		60° C.	
Per cent. of distillate below 100° C.	39·0	0·722	65·5	0·708	59·0	0·701	86·5	0·692	68·0	0·705	74·0	0·696
„ „ between 100° & 120° C.	49·0	0·748	26·5	0·742	28·5	0·736	11·5	0·739	23·0	0·743	15·5	0·736
„ „ between 120° & 133° C.	7·5	0·757	4·5	0·754	7·0	0·750	—	—	5·5	0·755	5·0	0·745
„ „ above 133° C. . . .	3·5	0·767	2·5	0·769	4·0	0·765	0·5	—	2·5	0·773	4·0	0·764
„ „ of sulphur . . .	0·03		0·06		—		0·06		—		0·06	
„ „ of loss . . .	1·5		1·0		1·5		1·5		1·0		1·5	
Calorific Value :												
B.Th.U. per lb. . . .	20,092		20,254		20,268		20,344		20,137		20,218	

Mixtures of petrol and benzene, and of petrol or benzene with less volatile fuels are often employed; Ballantyne considers that the great improvement in volatility and combustibility conferred upon less volatile fuels by the addition of about 20 per cent. of petrol or benzene has not been as fully recognised as it merits. In mixtures of this class he has occasionally found alcohol present. With such mixtures of petrol and less volatile fuels it is, however, important that the carburettor be adequately warmed in order to secure complete vaporisation of the fuel before it enters the cylinder; on this point some experiments by Mr. A. Duckham are of interest; in the *R.A.C. Journal,* June 4, 1908, p. 651, he states:

'Some time ago I carried out experiments which may be of interest in connection with the circumstances under which liquid, as distinguished from gaseous petrol, is carried into the cylinders. I had a car which I considered to be very inefficient in its consumption, and, wishing to ascertain whether any liquid, except water, could be condensed from the exhaust, I connected my silencer to a large well-cooled condensing coil, and found that I condensed a large quantity of a heavy spirit boiling approximately between 210° and 320° F., whereas the spirit in my tank boiled between 100° and 310° F. It immediately occurred to me that I was not using enough air, and that, as one would expect, a selective action was occurring, that portion of the petrol which was most easily vaporised having a preference in explosion, because of its intimate mixture in a gaseous condition with the air. I also fitted a glass induction pipe to the engine and found that a liquid was entering the cylinders, and that this liquid was partly in the form of a mist and partly in small streams on the sides, and principally on the bottom of the induction pipe. My next step, therefore, was to use a spirit boiling between 100° and 210° F., which I specially distilled for the purpose; using this I found that it entered the cylinders in a gaseous state, with practically no signs of mist or liquid on the sides of the induction pipe. At the exhaust I condensed practically no liquid, but the exhaust gases contained an alarming proportion of CO, and I had every reason to believe that there was a much greater deposit of soot on the valves in the case of this lighter spirit.

'My next experiment, therefore, was to largely reduce my jet, and at last I reached a point at which I obtained a minimum of CO at the exhaust and at the same time apparent freedom from sooting. In order to study the practical aspect of the question I made road tests, in the first case using the large ordinary-sized jet and the spirit boiling between 100° and 310° F., and in the second case with a smaller jet, using the spirit boiling between 100° and 210° F. In the first case I obtained between 21 and 22 miles per gallon, whilst in the latter case I could obtain 35, although it struck me at the time that, running at slow speeds, the engine did not do so well on this lighter spirit, and for this I cannot account. I may also add that I tried running the engine on the spirit boiling between 210° and 320°, using the smaller jet, and found that, *if I heated the petrol and the air to about* 100° *F.* above atmospheric temperatures, the mixture of petrol and air entered the cylinders free from mist and liquid, and apparently, after once getting her started, the results were quite good; a minimum of CO was found in the exhaust, and there was an absence of sooting.'

The practice of adding a small proportion (one-twentieth to one-fortieth) of lubricating oil to the petrol is sometimes adopted with a view to assisting valve and piston lubrication; the desired result is probably attained to some extent, though much of the oil spray suspended in the mass of the mixture within the combustion chamber is 'cracked' at the high temperature of explosion; the practice tends to favour carbon deposit and blue exhaust. Tarry deposits usually arise from excess of oil in conjunction with over-rich air-petrol mixtures, the finely divided carbon produced in part being deposited on the film of oil covering the piston head and cylinder surfaces and forming with it a tar-like substance. The black exhaust occasionally observed arises largely from imperfect evaporation of petrol; even if excess of air be present the existence of liquid spray in the cylinder will give the 'black' powdery exhaust effect.

Shale Petrol.—Most of the Scottish shale companies contribute supplies to the petrol market of this country. As early as 1898 Young's Paraffin Co. were supplying a shale spirit of 0·68 specific gravity under the name of 'Autoline'; of later spirits 0·725 'Dynol' has already been mentioned. In 1911 the principal Scotch companies supplying petrol were Young's Paraffin Co., The Oakbank Oil Co., J. Ross & Co., of Philipstown, N.B., and Clarkson, Thomson & Co., of Glasgow; this latter firm supply the 'Strathclyde' spirit.

Illuminating Oils, or Kerosenes.—'Kerosene' is an American name applied to the burning oils obtained by the distillation of crude petroleum, and which have a specific gravity ranging from about 0·78 to 0·83. Well-known brands are:

	App. sp. gr.
American 'Water White'	0·780
" 'White Rose'	0·784
" 'Royal Daylight'	0·797
Russian 'Russolene'	0·825

These kerosenes are now used also to some extent as fuel in oil engines; Russian oil is considered the best for this purpose, and a cheap standardised oil known as 'Rocklight' or 'R.V.O.' (Russian vaporising oil) is now largely supplied by the leading oil distributing companies; it has a specific gravity of 0·825 and a flash point of

86° F. by Abel close test (*v. infra*). For power purposes it is important that the oil used shall have no tendency to produce tarry deposits.

The kerosenes are mixtures of hydrocarbons mainly of the paraffin (C_nH_{2n+2}) and olefine (C_nH_{2n}) series; broadly, the American oils contain less olefines than the Russian; Prof. Mendeleff considers that the Russian and American oils contain the same hydrocarbons but in varying proportions; Sir B. Redwood ('Petroleum,' Vol. I, p. 232) says:

'Comparison of the results obtained with American, Galician, and Russian petroleum shows that the same classes of hydrocarbons . . . are present in the oil from all three sources, but that the relative amount of naphthenes' (isomers of the olefines) 'is greatest in Russian and least in American petroleum.'

The want of homogeneity of the kerosenes is well illustrated by some fractional distillation experiments of Prof. Robinson, which furnished the following results:

Description of oil	American kerosene 'White Rose.'	American 'Royal' Daylight.'	Russian kerosene 'Russolene.'
Specific gravity	0·784	0·797	0·825
Flashing point (Abel test) . .	105° F.	81° F.	88° F.
Vapour temperature at which distillation commences	293° F.	257° F.	239° F.
Distillates	Per cent.	Per cent.	Per cent.
Spirit below 300° F. . . .	7	23	20
Kerosene 300° to 520° F. . . .	85	54	68
Distillate above 520° F. . . .	5	10	9
Residue at 680° F.	3	13	3
	100	100	100

Thus it will be noted that 'White Rose' appears of fairly uniform composition, no less than 85 per cent. distilling between 300° F. and 520° F.; 'Royal Daylight' is much less uniform, while 'Russolene' occupies an intermediate position.

Prof. Robinson found also that the American 'Water White' burning oil, although of the low specific gravity 0·78, has the high flash point of 108° F.; it commences to distil only when its temperature reaches 347° F., and no less than 55 per cent. comes over between this temperature and 420° F., i.e. within the narrow range of 73° F.

He points out also that all the kerosenes may be easily vaporised without decomposition and without leaving any residue by simply blowing or bubbling dry air or dry steam through the liquid heated to about 500° F. in a distilling flask. Evaporation is also hastened by agitation, or spraying through a special nozzle, thereby exposing a large surface in a cloud of mist, which is inflammable.

PARAFFIN OILS

The burning oils obtained from Oil Shale are distinctively known commercially as 'Paraffin Oils,' although, as has already been mentioned, nearly all burning oils consist mainly of members of the chemical paraffin series, these preponderating in American kerosenes and being usually present in a somewhat less degree in Russian burning oils.

Oil shales occur in many countries; large deposits are found in Great Britain. Deposits occur also in Australia, France, New Zealand, Nova Scotia, Servia, and Spain. The Scotch oil shales are principally found in Midlothian and Lanarkshire; the seam at Broxburn, twelve miles west of Edinburgh, is one of the largest known, and is extensively worked, averaging 5½ feet in thickness.

Scotch oil shale is a rather glossy, blackish, flaky mineral; when rich in bituminous matter it can be cut, and if ignited will burn with a luminous flame; poor shales are hard and slaty in appearance.

The Scotch shale industry owes its creation to the energy of Dr. James Young, who, in 1847, experimented with a crude oil which exuded from the top of a coal seam at Alfreton in Derbyshire; from this he obtained a light burning oil, a heavy lubricating oil, and a quantity of paraffin wax. In 1850 Dr. Young discovered that by the slow distillation of bituminous coal and oil shale 'paraffin oil' could be obtained. The distillation of the shale is conducted in cast-iron retorts at a temperature of about 900° F., and results in the production of spent shale, crude shale oil, and ammonia liquor. Superheated steam is passed through the retort to carry over the oil vapours and ammonia, and prevent dissociation.

The ammonia liquor contains from 2 per cent. to 5 per cent. of ammonium carbonate and sulphide, and is utilised for the manufacture of ammonia suphate.

The crude shale oil is a green or brownish substance, solid at temperatures below about 80° F., and of specific gravity from 0·86 to 0·89. Broxburn shale averages a yield of 32 gallons of crude oil, 44 lbs. of sulphate of ammonia, and about 2000 cub. ft. of gas per ton. Inferior shales yield only from 16 to 20 gallons of crude oil, with a much larger quantity—60 to 70 lbs.—of ammonia sulphate, per ton. The spent shale is used as fuel for the retorts.

Crude shale oil consists principally of members of the paraffin and olefine series, and resembles many of the qualities of crude petroleum obtained direct from oil wells.

In the distillation of crude shale oil great care is exercised to prevent dissociation of the solid paraffins; superheated steam at a pressure of from 10 to 40 lbs. per sq. in. is blown through it to induce distillation at as low a temperature as possible. The various processes employed differ in detail, but the commercial products obtained are, roughly, as follows:

COMMERCIAL PRODUCTS FROM SCOTCH SHALE OIL. (*From Prof. Robinson*)

Name of product	Percentage by volume	Specific gravity
Gasolene & naphtha	3 to 6	0·665 to 0·730
'Paraffin oils' (burning)	30 " 40	0·790 " 0·820
Intermediate, or gas oils	4 " 10	0·850 " 0·880
Lubricating oils	12 " 18	about 0·885
Paraffin wax	10 " 12	0·87 to 0·91
Coke, gas, and loss	30 " 20	—

The burning oils are those used for many heavy oil engines, and they are divided into No. 1 burning oil (0·802–0·804), No. 2 burning oil (0·808–0·810), and lighthouse oil (0·810–0·820).

Flash Point and Fire Test.—Any mineral oil on being gradually heated at length evolves vapour which ignites with a momentary flash on application of a flame; the temperature at which this occurs is described as the flash point of the oil. On further heating a temperature is attained at which the oil itself ignites on flame application, and continues burning; this temperature is described as the fire test of the oil.

The flash point depends upon the way in which the experiment is made, i.e. whether by so-called 'open test,' in which the vessel containing the sample of oil is uncovered, or by 'close test,' in which it is covered during the experiment.

In the early days of the petroleum industry the ordinary specific gravity test prevented the introduction of an undue proportion of heavier hydrocarbons into burning oils, which would also have diminished their illuminating quality, but it did not prevent the addition of volatile hydrocarbons which might render the oil dangerous when used in ordinary lamps; in the interests of public safety legislation accordingly became necessary.

A very full account of the various instruments designed to determine exactly the flash point of oils is given by Sir B. Redwood ('Petroleum,' Vol. II, p. 546 *et seq.*). The problem, though apparently simple, has necessitated much attention and research ; the flash point is affected by the prevailing temperature, being considerably lower in tropical than in temperate climates ; it also differs materially at different barometric pressures, so that the altitude of the place of test affects the result. In 1875 the British Government requested the late Sir Frederick Abel to experimentally investigate the whole subject in order that it might be put on a satisfactory basis ; the result of his investigations was the legalisation in August 1879 of the 'Abel close test.'

Fig. 277 is a section of the Abel close test apparatus, from which it will be seen that a copper vessel C is provided which contains water marked W. This water forms a water bath. An air chamber, A, is placed within the water bath, and it carries within it an oil cup, P, made of gun metal. This cup rests upon an ebonite ring and over the chamber A, and has a tight-fitting lid on which is fixed a gas burner. The oil cup carries a thermometer, *t*, and above the cover is fixed a slide, which on being moved uncovers three holes. The gas jet, swivelling on a lever and moving with the slide, carries a small flame, and the arrangement is such that, as the lever tilts, the flame is passed through one of the openings in the slide and reaches the surface of the oil in the cup.

The thermometer *t′* indicates the temperature of the water bath, and the spirit lamp *b* supplies the necessary heat. A pendulum 24 ins. long times the operation of testing the flash.

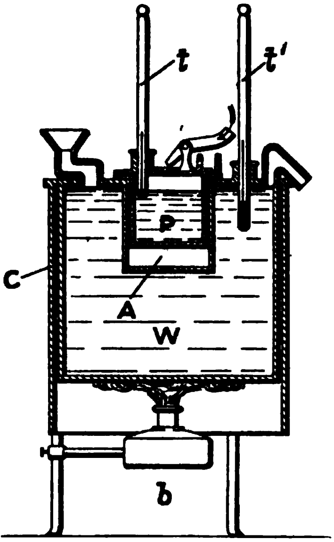

FIG. 277

To determine the flashing point of the oil, the temperature of the water bath at the start of the test is arranged at exactly 130° F. The oil to be tested is cooled to 60° F. and poured carefully into the oil cup P, avoiding splashing, until the oil reaches the point of a small bent wire gauge inside the cup. The lid is then put on, and the cup placed in the bath, the rise of the temperature being watched on the thermometer *t* in the petroleum cup. When the oil reaches a temperature of 66° F. the testing is started by setting the pendulum in motion, and while it makes three oscillations drawing the slide slowly open, and at the fourth oscillation closing it rapidly. By this the test flame is gently tilted through a hole in the slide to the space above the oil. This operation is repeated once for every increase of temperature of 1° F. until the vapour of the oil ignites within the oil cup, giving a pale blue flicker or flash. The temperature of the oil at which this occurs is the flashing point.

The lowest flashing point allowed by law for petroleum intended for burning in lamps in this country is 73° F. or 22·8° C. It is very important, therefore, in experimenting upon various samples of oil, to be certain that the oil is above the legal flashing point.

A model Abel apparatus is deposited in the Weights and Measures Department of the Board of Trade.

Correction for Barometric Height.—The flash point is found on the average to vary by about 1·6° F. per in. of barometric height, rising when the mercury rises, and *vice versa*; as the standard barometric height for testing is 30 ins., a correction accordingly becomes necessary. For example, if the observed flash point be 70° F. with barometer at 28 ins., the legal flash point is $70 + (2 \times 1{\cdot}6) = 73{\cdot}2°$ F., whereas had the barometer stood at 31 ins. the flash point would be reckoned as $70 - (1 \times 1{\cdot}6) = 68{\cdot}4°$ F., and the oil would not be in conformity with the Regulations under the Act.

The Abel-Pensky Tester.—In the course of investigations conducted in 1880 on behalf of the German Government it was considered that of the many types of flash point testers in use the Abel was the best; but exception was taken to the method of applying the test flame, and in Pensky's improved apparatus the motion of the slide and application of the flame are effected by a clockwork mechanism, thus eliminating altogether the personal element in the test. As conducted in Germany, tests made with the Abel-Pensky instrument furnish results about 3° F. higher than those given by the British standard 'Abel' tester.

In 1881 Messrs. Engler & Haass concluded from a number of experiments on flash point testers that it was desirable to have a stirrer in the oil cup, as a stratum of vapour was liable to form on the oil surface and the temperature of the oil to be not uniform throughout its mass. For the testing of kerosenes, however, it appears in practice to be unnecessary to make this addition, though when the flash point of lubricating oils and heavy oils is required it becomes desirable.

Thus in the Pensky-Martens tester, much used in tests of the heavier oils, a stirrer consisting of two sets of small paddles, one in the oil and one in the vapourous space above, is caused to rotate by means of a flexible wire tube; the oil cup is heated in an iron bath containing air and provided also with an air jacket to reduce loss of heat by radiation, by means of a Bunsen flame. By removing the cover of the oil cup the apparatus may be used for making *open* tests; in this way the flash point and fire test may be determined as in United States practice, the oil cup being directly heated at a rate of 8° F. per minute and the test flame not exceeding ¼ in. in diameter.

Test-tube Method.—Sir B. Redwood ('Petroleum,' Vol. II, p. 587) states that when a small quantity only of oil is available for testing, the flash point may be found by slowly heating the sample in a test-tube about ¾ in. in diameter and 4 or 5 ins. in length; the tube should be filled with oil to the extent of about one-third, and a delicate thermometer employed to obtain its temperature. After agitating the oil the test flame is applied by the insertion of the burning end of a piece of thin twine into the mouth of the tube at intervals. He says:

'In this way results which do not differ materially from those furnished by the Abel instrument may be obtained after a little practice.'

The flashing point of an oil determined by an 'open' test is commonly from 15° F. to 20° F. *higher* than as determined by the standard British Abel close test. The fire-test is from 40° F. to 50° F. higher than the Abel close test, as illustrated by the following figures for American kerosene:

Fire-test	110° F.	120° F.
Abel close test	70° F.	73° F.

In the table on p. 462, prepared from some of Professor Robinson's results, some physical properties of typical kerosenes, together with a number of crude, intermediate, and residual oils, are given. It will be observed that, excepting the last three cases, the percentage composition varies but little from an average of 85½ per cent. carbon, 13½ per cent. hydrogen, and 1 per cent. oxygen; the calorific values, determined by a Berthelot bomb calorimeter, show also a considerable degree of constancy, the mean value of 19,600 B.Th.U. per lb. being but little departed from. The calorific values of 'petrol' from Thomas & Watson's experiments, using a Boys calorimeter, are given on p. 452 and those of Blount with a bomb on p. 453; Professor Robinson, using the bomb type, obtained the value 19,800 for petrol of 0·678 specific gravity at 60° F.; it has already been pointed out that the results obtained with the bomb calorimeter appear to be somewhat higher than when certain other types are employed.

In the table on p. 463 further useful particulars of a selection of well-known kerosenes and 'intermediate' oils, also from Professor Robinson's results, are given.

Residual Oils.—The substances remaining after distillation of the

SOME PHYSICAL PROPERTIES OF KEROSENES, CRUDE AND RESIDUAL OILS

(*From Prof. Robinson's Results*)

Description of product	Origin	Sp. gr.	Per cent. chemical composition.			Calorific value, B.Th.U. per lb.	—
			Carbon	Hydrogen	Oxygen (by difference)		
Petroleum refuse . .	Baku	0·928	87·1	11·7	1·2	19,260	
Ostatki	Baku	0·906	84·9	14·0	1·1	18,612	
Light crude oil . .	Baku	0·884	86·3	13·6	0·1	20,628	
Heavy crude oil . .	Baku	0·938	86·6	12·3	1·1	19,440	
Refined oil 'Russolene' .	Baku	0·825	86·0	14·0	0	20,286	Pale straw colour
Refined 'Royal Daylight'	N. America	0·797	85·7	14·2	0·1	20,100	Pale straw colour
Heavy crude oil . .	Pennsylvania	0·886	84·9	13·7	1·4	19,210	
Heavy crude oil . .	America	—	86·9	13·1	0	19.642	
'Refined petroleum' .	America	—	85·8	14·2	0·3	19,885	
'Double refined' . .	America	—	80·6	15·1	4·3	19,955	
Pratt's Motor Spirit .	America	0·719 (60° F.)	85·2	14·8	0	18,610	(From Thomas & Watson's experiments)
Solid residuum . .	America	—	97·9	0·5	1·6	14,500	Practically pure coke
Blast furnace oil . .	Scotland	0·920	83·6	10·6	5·8	16,080	
Crude coal tar . .	Gas-works	1·05	82·0	7·6	10·4	16,050	

SOME PROPERTIES OF KEROSENES AND INTERMEDIATE OILS

(*From Prof. Robinson*)

Name of oil	Colour	Specific gravity at 15·5° C. (60° F.)	Specific gravity corrections per 1° C.	Coefficient of expansion per 1° C.	Specific heat	Flashing point by Abel close test		Boiling-point of liquid	Distillation.			
						Fah.	Cent.		Volume distilled under 215° C. (liquid)	Highest temperature (liquid)	Volume distilled	Time
Burning Oils :						deg.	deg.	° C.	Per cent.	° C.	Per cent.	hours
American Royal Daylight	Light straw	0·811	0·00067	0·00084	0·47	76	24·5	144	25	230	35	3
,, ordinary	,, ,,	0·791	—	—	—	75	24	145	29	223	56	3
,, Water White	Colourless	0·780	—	—	—	108	42	150	55	216	55	4
,, Tea Rose	Light straw	0·797	—	—	—	83	28·3	150	22	243	37	3
Russian ordinary (Russolene)	,, ,,	0·824	0·00068	0·00085	0·45	82	27·8	151	30	221	36	3
Russian Lustre	,, ,,	0·825	0·00072	0·00089	0·45	—	—	—	—	—	—	—
Broxburn Lighthouse	,, ,,	0·810	0·00072	0·00089	0·44	152	66·7	165	1st drop.	243 270 300	55 90 100	3 2 3
Intermediate Oils :												
American mineral sperm	Straw	0·833	—	—	—	—	—	195	0	300	5	3
Storrar's Scotch gas oil	Reddish brown	0·843	—	—	—	—	—	195	0	283	5	3
Scotch intermediate shale oil	Clear brown	0·846	—	—	—	—	—	195	0	291	18	2
Light lubricating oil	,, ,,	0·853	0·00068	0·00080	—	225	107	195	0	285	18	2

ethers, naphthas, kerosenes, gas oils, and lubricating oils are known by the general name of 'residuum' in America, and 'ostatki' (i.e. dregs) in Russia. These residua differ much in appearance and properties, according to the earlier treatment of the crude oil; thus in the early days it was common to distil the crude oil almost to dryness, the process being only stopped short of 'coking'; in Russia, however, there is a large demand for ostatki as a liquid fuel, and the treatment of the crude oil is accordingly, in general, not so exhaustive as in American practice.

Residuum is sometimes fluid, sometimes a viscous, semi-solid, dark-green or dark-brown liquid with an unpleasant empyreumatic smell; the thick, easily-solidified residua often contain paraffin wax, though tarry substances are sometimes the cause of their viscidity. The more fluid residua usually contain no paraffin.

Ostatki is used for heating the stills in refineries and for steam-raising purposes generally; also sometimes as a lubricant for rough machinery after separation of water, removal of the unpleasant smell, and improvement in colour by treatment with sulphuric acid. The most usual method of burning liquid fuel in practice is to spray it through an injector burner by aid of superheated steam into a fire-brick lined combustion chamber; the oil is thus heated and vaporised. The fire-brick lining protects the plates from the direct action of the flame, and acts as a reservoir of heat, maintaining a high temperature within the fire-box.

In addition to petroleum refuse there are in Great Britain other cheap residual oils obtained from coal-fed blast furnaces, coke-ovens, and gas works, which are used also as liquid fuel. These are commonly known as blast furnace oil, creosote oil, coal-gas tar, green oil, and oil-gas tar. Of all the liquid fuels, however, Russian ostatki of specific gravity about 0·9 is the best; it has a flash point of about 280° F. and a heating value of about 19,000 B.Th.U. per lb. The best performance, obtained with a new railway locomotive, has shown an evaporation of 14 lbs. of water to steam at 140 lbs. per sq. in. pressure per lb. of Ostatki, corresponding to the very high boiler efficiency of 86 per cent.; broadly, 1 lb. of Ostatki is as effective as 2 lbs. of coal.

Where the supply of liquid fuel is adequate and regular, and the price low, it is largely and increasingly used for firing stationary, locomotive, and marine boilers.

Diesel Fuel Oils.—The heavy crude and residual oils best suited for Diesel engines have a specific gravity between 0·85 and 0·92 and a lower calorific value between 18,000 and 19,000 B.Th.U. per lb. Dr. Diesel defines suitable oils of this type as crude mineral oils having a hydrogen content of over 10 per cent. by weight, freed from benzine, without solid impurities and having a calorific power of more than 18,000 B.Th.U. per lb. Suitable lignite tar oils should have a similar hydrogen content, and have a calorific value of over 17,000 B.Th.U. per lb. Dr. Diesel also states that oils from animal and vegetable sources may be used; he has successfully operated a Diesel engine with earth nut oil having 11·8 per cent. of hydrogen and a calorific power of 15,400 B.Th.U. per lb.

Methods of Vaporising and Decomposing.—Before discussing the vaporisers in actual use, it is advisable to consider some of the laboratory methods of vaporising, in view of the difficulty of providing vaporisers which will treat varying oils of high flashing point and density.

When a homogeneous substance like water is boiled, the temperature remains constant from the moment of boiling to the complete distillation of the whole liquid.

Likewise if dry air be blown through water, every cubic foot of air will carry off a certain volume of water vapour, until the whole of the water is evaporated, and this will occur by blowing through air at any temperature at which water has an appreciable vapour tension.

The *vapour tension* of water is the pressure of water vapour at any given temperature. The term vapour tension is generally used for pressures under atmospheric pressure.

The following table gives the vapour tension of water for different temperatures from 0° C. to 100° C. The tension is given in millimetres mercury; that is, the tension of the water vapour at each temperature is given in the height of mercury column which the particular pressure of water vapour at that temperature is capable of supporting.

VAPOUR TENSION OF WATER VAPOUR

Temp. C.	Tension mm. mercury	Temp. C.	Tension mm. mercury
0°	4·6	40°	54·91
5°	5·53	50°	91·98
10°	9·17	60°	148·70
15°	12·70	70°	233·09
20°	17·39	80°	288·51
25°	23·55	90°	525·45
30°	31·55	100°	760·00

From this table it will be observed that at 15° C., about the ordinary temperature of the atmosphere, the tension or pressure of water vapour is equal to 12·7 mm. mercury. The total pressure of the atmosphere is taken as 760 mm. mercury, from which it would appear that the pressure of water vapour at that temperature is about $\frac{1}{60}$ of the pressure of the atmosphere, so that if water were to be evaporated by passing dry air through it at that temperature, 60 cub. ft. would require to be passed through to take away 1 cub. ft. of water vapour, that is to take away a volume of vapour sufficient to make 1 cub. ft. of steam supposed to be at atmospheric pressure and temperature. If, however, the temperature be raised to about 80° C., 2 cub. ft. of dry air would carry away about 1 cub. ft. of steam calculated at atmospheric pressure.

Water can thus be evaporated either by boiling it off by raising the temperature above the boiling-point, or by passing air through it or any other gas at a temperature below the boiling-point; and the amount carried off by a cubic foot of air depends upon the temperature of the water.

The important point to remember is, that to however low a temperature the water be reduced, it can be entirely evaporated by treatment with a sufficient volume of air.

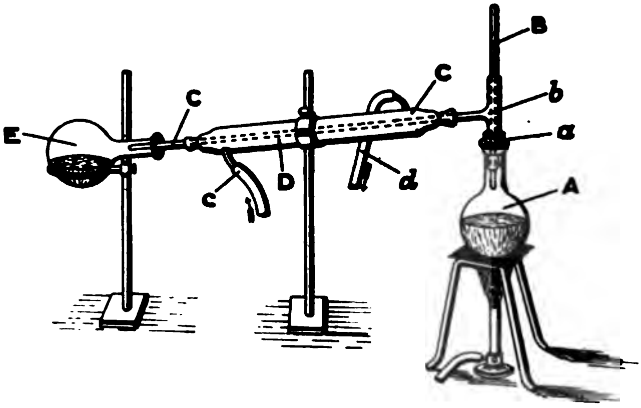

FIG. 278

Petroleum or oil in the same way can be evaporated either by boiling off, or by treatment with air or steam; and the temperature at which the whole liquid can be evaporated is much reduced by passing hot air through or over the liquid, instead of attempting to boil the liquid away. Thus many of the American oils, which leave a considerable residue at 358° C., can easily be evaporated by passing hot air through the liquid, without requiring any further rise of temperature. It is often objectionable to attempt to vaporise by boiling off or distilling, because in many oils the boiling-point is so high that the decomposition point is reached before the liquid will boil. In such a case, attempting to force vaporisation or distillation by increasing the heat only results in the chemical decomposition of the oil, and leaving in the vaporiser a comparatively large quantity of carbon or tar. A sample, for example, of solid paraffin, such as is used for candles, could not be entirely distilled by any attempt at boiling; but if the sample be placed in a vessel, which is heated to the highest temperature the paraffin will stand without decomposition on a sand bath—say about 400° C.—and hot air or superheated steam be blown through the liquefied paraffin, then nearly the whole of it can be distilled without decomposition. From this it follows that, if vaporisation is desired without decomposition, the temperature can be kept much lower by heating the vaporiser to a predetermined point, and then passing hot air through or over the liquid contained in it.

It is interesting to note, in connection with the decomposition of paraffin and olefines by heat, that mere heating up in a closed vessel does not produce any large amount of decomposition. If, however, the oil or paraffin be heated up under pressure in such manner that the ordinary boiling-point is considerably exceeded, and that oil be distilled and condensed in a condenser—also under pressure—then the oil rapidly decomposes.

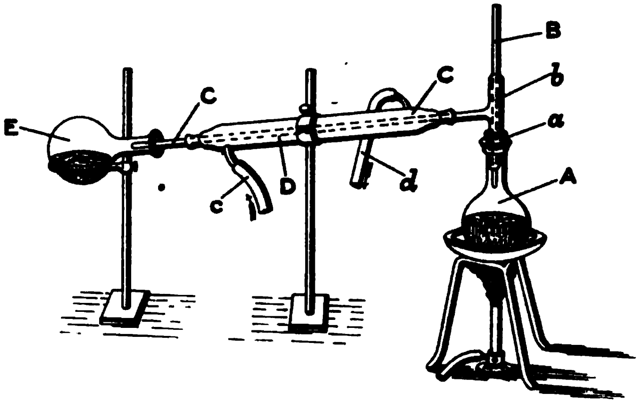

Fig. 279

Some well-known laboratory methods of experimenting illustrate in a vivid manner the various facts which are useful to the engineer designing oil engines. The distillation of water, for example, in the laboratory apparatus shown in fig. 278, and the subsequent distillation of oils in the same apparatus, enables one to realise the difference between the nature of oils and water.

The apparatus is very simple, and consists of a glass flask A having a tightly fitting cork *a*, through which passes a glass T piece *b*, carrying the thermometer B. The free end of the T piece slips into the glass condenser tube C. This condenser tube passes within a water jacket tube D, fed with a current of cold water by the side tube *c*, which current discharges at *d*. The condenser tube terminates in the glass receiving flask E, supported upon a retort stand; the condenser is held by a clamp, also supported on a retort stand, and the distilling flask rests upon wire gauze supported on a tripod, and is heated bv a Bunsen flame.

It is an interesting exercise to rig up this apparatus, and distil fresh water from the flask, observing the thermometer during the process. Fresh water will boil away to the last drop, and collect in the receiving flask, while the thermometer remains steady at 100° C. from the beginning of the boiling to the completion of the distillation.

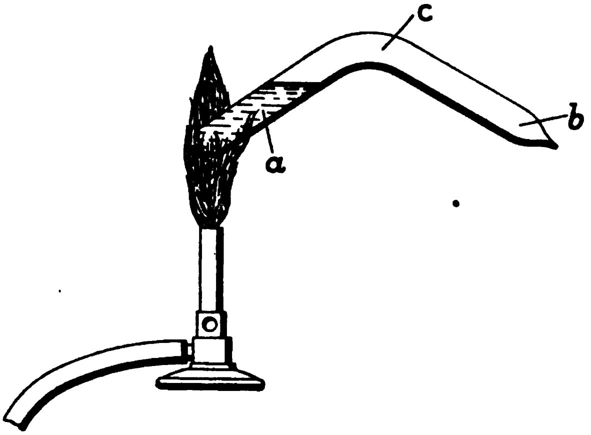

Fig. 280

If a sample of Royal Daylight oil be placed in the distilling flask (carefully dried from water), it will be found that the oil begins to boil about 144° C., that a lighter oil first passes over, and that the thermometer slowly rises, so that at 340° C. only 82 per cent. of the whole has distilled over, and even at 358° C. a considerable liquid residue will be left in the vessel. If the receiving flask be frequently changed in the course of the distillation, oils of different densities will be collected, the lighter oils boiling off first, and the heavier in order later. Such a process of distillation is called fractional distillation, and on the manufacturing scale it is practised to purify the oils, and separate the light from the heavy. In making this experiment with oil, the apparatus should be modified as shown in fig. 279, where the wire gauze is replaced by a sand bath, in order to protect the glass flask containing oil from the direct action of the flame In distilling oils experimentally from glass flasks, it is well to limit the size of the flask not to exceed 250 c.c. (quarter litre); and a quantity of dry sand should be kept at hand to extinguish the oil flame if the flask breaks and ignites.

It is found that as the lighter oils distil off and the thermometer rises, the oil in the distilling flask gradually becomes darker in colour, and at the high temperature of 350° C. it becomes quite brown. At first it is of a pale straw colour, and this change to brown proves chemical decomposition to be going on.

If a quantity of the oil which refuses to boil at even the high temperature of 350° C. be placed in one end of a bent glass tube, C, fig. 280, and the tube sealed up by the blowpipe flame, and the liquid distilled from the end *a* into the end *b* without applying any cooling, but after distilling returned again to the end *a* and distilled to *b* again; the process being repeated say for about twelve times; it will then be found on opening the glass tube that the oil subjected to this distillation under pressure has changed its nature very considerably. This can easily be proved by returning it to the flask A, fig. 279, and testing the boiling-point. It will be found that the liquid which before refused to boil at 358° C. will now begin to boil below 140° C., and the greater part of it will distil over long before 300° C. is reached.

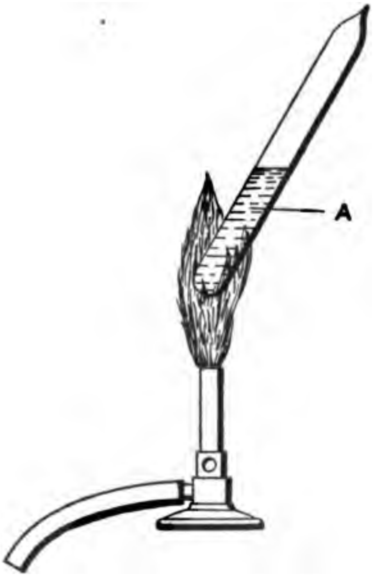

FIG. 281

A sample of the same heavy oil remaining from the first oil experiment if placed in a straight sealed tube as A, fig. 281, may be heated and cooled to the same extent as and for the same time as with the bent tube in fig. 280, and after these series of heatings and coolings it will be found to have hardly changed its composition. These oils if merely heated under pressure without distillation can bear comparatively high temperatures without decomposition, but if distilled at high temperature decomposition results.

This appears due to the recombination of the oils when heated to a high temperature and cooled slowly. For effective decomposition it is necessary to distil.

American petroleum refiners treat the heavy oil remaining after removal of the spirits and kerosenes by this process, which is termed '*cracking.*'

By the cracking process a considerable proportion of this heavy oil may be converted into lighter and more stable oils of lower boiling-point which are suitable as illuminants, thus increasing the yield of kerosene and even lighter grades. It is effected (*a*) by distillation under pressure, and therefore at a temperature above the normal boiling-point of the liquid; (*b*) by so constructing the still that the liquid condensed in the relatively cool upper part trickles back into the highly heated residual liquid, thus causing its dissociation into lighter compounds of lower boiling-point, which then distil over and collect in the condenser. During the cracking a considerable quantity of permanent gas is given off which is utilised to assist in heating the still. The process is very extensively employed. According to Redwood the action probably consists broadly in the reduction of a heavy paraffin by heating under pressure into a lighter paraffin and an olefine, somewhat as indicated by the formula:

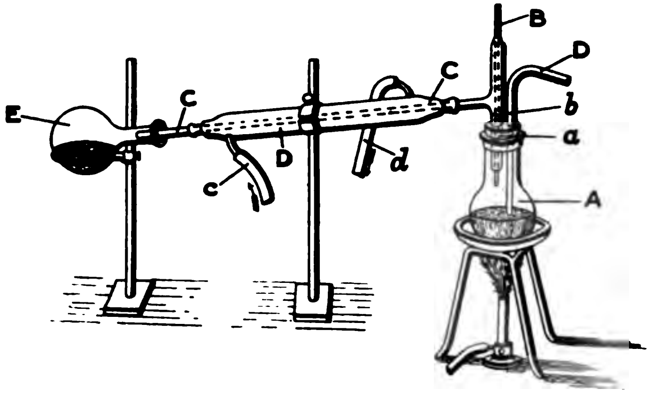

FIG. 282

$$C_nH_{2n+2} = C_{n-x}H_{2(n-x)+2} + C_xH_{2x}.$$

Heavy paraffin = Light paraffin + Olefine.

Actually a mixture of several paraffins and olefines is produced; the result obtained depends entirely upon the temperature and treatment adopted. For example, by employing a sufficiently high temperature methane (CH_4) can be driven off, and coke alone left in the retort.

If the thermometer be removed from the distilling flask, fig. 279, before the temperature rises so high as to damage it, and the heat be further raised, it is found that after a time a tarry mass is left in the flask which cannot be removed by heating. These experiments very clearly show that the particular oil could not be vaporised by boiling off without leaving a considerable residue. It would, therefore, be hopeless with this oil to design a vaporiser to boil off the oil as vapour, it would only result in the vaporiser being choked with tar and carbon deposit in a few hours.

Some method is therefore required which will vaporise the whole of this heterogeneous oil, the heavy part as well as the light. This can be done in another way by means of the apparatus shown in fig. 282, which is the same as that shown in fig. 279 except that the flask A has a wider neck, and the cork carries in addition to the T piece and thermometer the air tube D. If the flask A be charged with Daylight oil and heated up to about 140°, and air be then slowly bubbled through the oil (from a gasometer), it will be found that almost the whole of the oil can be distilled out of the flask A without leaving any heavy residue, and the temperature of the thermometer need not be raised above about 200° C. The contents of the flask will pass over without decomposition and without leaving any clogging residue or carrying over any tarry matter.

If a sample of solid paraffin be placed in the flask fig. 282 and heated up to about 350°, and dry steam be then blown through by the pipe D, it will be found that even solid paraffin will distil over practically without decomposition.

If the paraffin be heated highly alone and distillation attempted, it rapidly decomposes, leaving a charred carbon mass.

From these experiments it is evident that the best method of vaporising a hydrocarbon oil containing heavy as well as light hydrocarbons is to heat the oil in a vaporiser to a moderate temperature, say about 300° C., and then pass air through or over it also heated to about the same temperature. By treating it in this way the whole of the oil, light and heavy, can be vaporised without fear of decomposing the oil and so producing tarry products or carbon residues.

It is a mistake to use red-hot surfaces in vaporising an oil when the vapour formed has to pass through valves ; it is a mistake, however, which inventors often make.

An oil like 'Broxburn Lighthouse,' boiling entirely below 300° C., might be treated in another way, but the method described of passing hot air through would easily vaporise it also, so that no other method is necessary.

The Vacuum Process.—Boiling under reduced pressure supplies a means of distilling oil at a comparatively low temperature, and lubricating oils are now largely obtained by the so-called 'vacuum process.' A simple direct-fired vacuum still is diagrammatically shown in fig. 283. The oil to be treated is contained in the vessel A, from which the vapours pass through the worm condenser B and collect in the receiver C. A Koerting ejector D, supplied with steam from the small pipe E, exhausts any air, steam, or other vapours from the still, coil, and receiver, and discharges these through the pipe F; the condenser distillate is drawn off at intervals at the cock H.

Viscosity of Oils.—The viscosity of a mineral lubricating oil is a most important point to determine, and this is arbitrarily estimated by means of instruments termed viscometers. In Great Britain the Redwood viscometer is generally employed; in the U.S. the Saybolt; while in Germany the Engler type prevails. A diagrammatic section of the Redwood viscometer is shown in fig. 284. It consists of a silvered copper oil cylinder, A, $1\frac{7}{8}$ ins. in diameter and $3\frac{1}{2}$ ins. in depth, having in the centre of the bottom an agate jet, J, fitted into a slightly conical metal seating. This cylinder is fixed in a copper water bath, C, having a heating tube, E, projecting downwards at 45°; in this water bath is a revolving paddle, H, carrying a curved shield, K, to prevent splashing; in this shield is fixed a thermometer, T, to indicate the temperature of the liquid in the water bath.

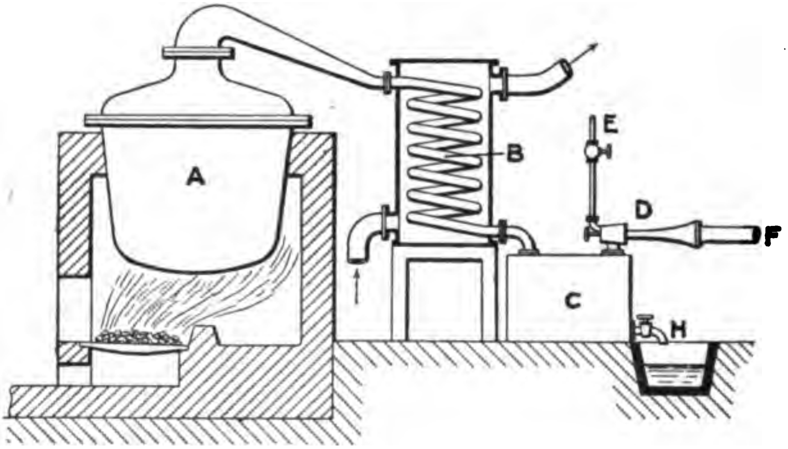

FIG. 283

The oil-cylinder A has a stopper rod, V, terminating in a small brass sphere which rests in a hemispherical cavity in the agate jet. A thermometer, T_1, indicates the temperature of the oil. Within the oil cylinder is fixed a small upturned point, B, which serves as a gauge of the height to which the cylinder is filled. The instrument is supported on three levelling screws.

Great care is taken to secure uniformity in the preparation of the agate jets, and the instruments are all standardised. To make a test, the water bath is first filled with a suitable liquid to a height equal to that of the point B in the inner cylinder; for temperatures up to 200° F. water may be used; for higher temperatures a heavy mineral oil is a suitable liquid.

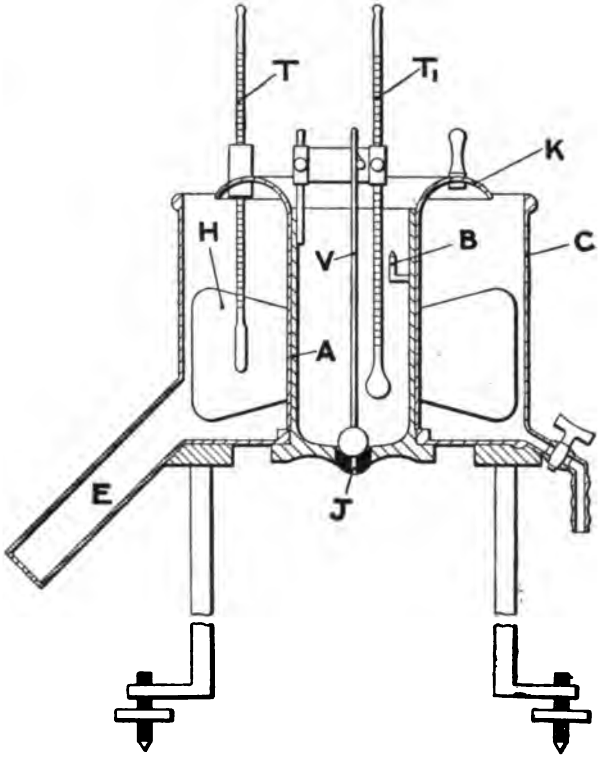

FIG. 284

This liquid being brought to the required temperature, the oil to be tested, which may be previously heated to the same temperature, is poured into the oil cylinder A until its level just reaches the point B. A narrow-necked flask holding 50 c.c. to a marked point is now placed beneath the jet J, in a vessel containing a liquid at the same temperature as the oil. The ball valve V is then raised, a stop watch simultaneously started, and the number of seconds occupied by the outflow of 50 c.c. of oil noted.

When tests are made at a temperature very much above the normal a gas flame is applied to the heating tube E and the paddle gently rotated during the experiment in order to maintain the temperature of the bath uniform. Care should be taken to place the instrument level, and that the oil under test be free from dirt, water, or other impurity, and also that it shall not have been heated above the temperature of the test during the preceding twenty-four hours.

Very commonly, in commercial testing, the viscosity is estimated by the number of seconds occupied by 50 c.c. of the sample in passing through the agate jet; sometimes also the viscosity is stated relatively to that of water at 60° F. regarded as unity. In the following short table a few values are given of viscosities relatively to water at 60° F. as unity which may be useful for reference:

Description of oil	Specific gravity at 60° F.	Relative viscosity; water at 60° F. = 1				
		At 70° F.	At 120° F.	At 180° F.	At 300° F.	At 350° F.
Cod liver oil	—	7·9	3·0	—	—	—
Sperm oil	0·879	5·5	2·5	1·62	1·18	1·10
Refined rape oil	0·915	12·2	4·1	—	—	—
Light American machine oil	0·905	8·8	2·5	—	—	—
Light Russian ,, ,,	0·908	43·3	6·5	—	—	—
Price's 'Heavy gas engine oil'	0·906	39·8	7·84	2·47	1·31	1·20
'Motorine A'	0·893	146·5	24·2	5·88	1·74	1·52
'Motorine B'	0·894	103·0	18·6	4·82	1·62	1·48
'Huile D'	0·899	83·4	14·2	3·52	1·44	1·30
Light cyl. oil (American)	0·925	—	—	—	1·39	—
Dark cyl. oil ,,	0·905	—	—	—	1·76	—
Valvoline (cyl. oil)	0·949	—	—	—	1·57	—
American 'Royal Daylight'	0·797	1·165	—	—	—	—

ALCOHOL

Much attention was devoted between 1898 and 1904 to the use of industrial alcohol as a fuel, both alone, and mixed with from 10 per cent. to 50 per cent. of petrol, the German Government in particular, probably from politico-economical considerations, offering great encouragement; as a result the price of alcohol per gallon fell from about three shillings and ninepence in 1887 to ninepence only (in 1500 gallon lots) by 1904, in that country. In France also the use of alcohol and petrol-alcohol mixtures has been encouraged, and public competitions of automobiles using these fuels have been held on many occasions. The first of these, organised by 'Le Vélo,' took place in

April 1899, but was not a success, mainly through very adverse weather; the second 'Le Vélo' competition was held in October 1900, when 62 vehicles entered, 50 started, and 30 finished; among the entrants were included such well-known firms as Panhard, Clément, Gobron-Brillié, Rochet-Schneider, de Dietrich, Georges-Richard, Peugeot, and Darracq. In most cases the ordinary petrol arrangements were retained, but a few competitors enlarged their carburettors, inlet pipes, and valves; the fuels used were generally petrol-alcohol mixtures containing from 25 per cent. to 60 per cent. of petrol; four competitors used alcohol alone; the total length of the course was 76 miles.

In April 1909 the Paris-Roubaix trials (about 167 miles) took place; in this competition 83 entrants appeared; these were grouped in three classes, viz.:

1. Those using alcohol alone.
2. Those using more than 75 per cent. of alcohol.
3. Those using more than 50 per cent. of alcohol.

Further trials took place in July 1901 (Paris-Braisne), and in October 1901 a competition was held under the patronage of the French Minister of Agriculture. In connection with this last competition bench tests of an Aster and a Panhard motor were also conducted; the general result arrived at was that the same amount of mechanical work could be obtained from 1 litre of petrol as from 1·78 litres of alcohol.

Again in July 1903 an alcohol-petrol competition organised by the Automobile Club of France took place; the results of this are given in Vol. VIII of the *Automotor*, p. 887. In connection with this competition kerosene was tried in some cases, and found to act as a 'carburant'; but this was not a new discovery, as it had long been known that any good spraying carburettor would enable a motor engine when hot to be run with this fuel; the practical difficulties in regular use arising from the formation of carbon and tarry deposits, pre-ignition troubles, easy upset of 'mixture,' very smoky exhaust, &c., &c., still remain as serious obstacles to its employment as a fuel for the engines of road vehicles. In Great Britain the alcohol problem has not yet attracted serious attention; it is certain that very great difficulty would be experienced in obtaining the necessary legislative changes to enable it to be produced at a price at which it could be profitably employed.

There are two kinds of alcohol made for industrial purposes, viz.:

(1) Ethyl alcohol, C_2H_6O, the more ordinary form, which is obtained by the fermentation of various starchy and sugary substances as potatoes, rye, corn, sugar beet, sugar cane, &c.

(2) Methyl alcohol, $C_2H_4O_2$, known also as wood alcohol, obtained by the dry distillation of wood in iron retorts; it is a liquid with a somewhat strong odour. To render ethyl alcohol undrinkable it is usual to mix with it unpalatable substances, such as methyl alcohol (whence the name 'Methylated spirit'), benzol, acetone, &c.; the table on p. 476 gives the composition of some commercial alcohols.

In the French tests of October 1901 the 'Alcöol denaturé' used was a mixture consisting of 100 litres of ethyl alcohol (90 per cent.), 10 litres of methyl alcohol (90 per cent.) containing a small proportion of acetone, and 0·5 litre of heavy benzene; the specific gravity of this mixture at 15° C. was 0·833.

COMPOSITION OF COMMERCIAL ALCOHOLS. (*From Carpenter & Diederichs*)

Country	Hydrated ethyl alcohol	Impure methyl alcohol	Acetone	Pyridine	Benzol	Benzene	Sp. gr. of product at 15° C.
	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	
France . . .	89·5	7·5	2·5	—	—	0·5	0·832
Germany:							
'Denatured' .	97·5	1·5	0·5	0·5	—	—	0·819
'Motor' . .	96·75	0·75	0·25	0·25	2·0	—	0·825
Austria:							
'Denatured' .	94·5	3·75	1·25	0·5	—	—	0·835
'Motor' . .	97·0	0·3	Trace	Trace	2·5	—	0·826
Russia . . .	84·5	10·0	5·0	0·5	—	—	0·836
Italy (Motor) . .	89·85	6·5	2·0	0·65	1·0	—	0·835

The fuel used under the name 'Electrine' was composed of fifty volumes of the 'Alcöol denaturé' and fifty volumes of a carburant chiefly consisting of '90 per cent. benzol,' the remainder consisting of lighter hydrocarbons; the specific gravity of this mixture at 15° C. was 0·835.

Alcohol has a great affinity for water, and industrial alcohols contain this in various proportions.

Alcohol engines are now numerous, particularly in Germany, and very high thermal efficiencies are obtained in some instances; thus, of ten alcohol engines which competed for a prize offered by the German Agricultural Society in 1902 no fewer than three showed efficiencies varying from 30·9 per cent. to 32·7 per cent. on full load. The compression pressure employed was considerably higher than that usual in petrol engines, rising in some instances to 190 lbs. per sq. in.; with a maximum explosion pressure of roundly 500 lbs. per sq. in. Dr. Ormandy attributes the high efficiency to the high compression and to the effect of the 10 per cent. by volume of water contained in the alcohol used, in keeping down the temperature of compression. Dr. Ormandy[1] further states that many German alcohol engines were built between 1897 and 1901 which consumed only from 0·85 to 0·95 lb. of alcohol per HP hour.

To obtain best results the engines must be specially designed; larger carburettors, inlet pipes, and valves are required than for petrol; as the latent heat of alcohol much exceeds that of petrol, and also on account of the water present, the carburettor must be well jacketed, and the incoming air should be heated, preferably to about 190° C., to ensure complete vaporisation. The cylinder jacket water should also be kept nearly at boiling-point; too cool a cylinder results in imperfect combustion, acetic acid, aldehyde, &c., being produced instead of CO_2 and water; this involves loss of efficiency and corrosion of the internal parts of the engine.

Starting from cold with alcohol has long been a problem of difficulty. Alcohol engines are often arranged to start on petrol; when hot, the petrol is switched off and the alcohol—supplied by a separate exhaust-heated carburettor—switched on. In some instances this is done automatically, the engine always both starting *and stopping* on petrol, and thus being free from alcohol combustion products when not running; or, again, the engines are run on a mixture of alcohol and benzol containing from 10 per cent. to 50 per cent. of the latter 'carburant.'

The following test results obtained by Prof. E. Meyer in 1901 on a 14 HP 'Locomobile' alcohol engine of 8·28 ins. bore and 11·8 ins. stroke, with a volume ratio of compression of 5·91, are of interest in this connection:

PROF. E. MEYER'S TESTS OF A 14 HP ALCOHOL-BENZOL ENGINE; NORMAL FULL LOAD

Fuel used; per cent. by weight			B.Th.U. per lb.	Revs. per minute	BHP	Fuel per BHP hour, lbs.	Per cent. efficiency
Alcohol	Benzo.	Water					
87·2	—	12·8	10,440	280·3	13·91	1·00	24·3
86·4	—	13·6	10,350	273·3	13·6	1·03	23·8
90·9	9·1	—	10,980	282·0	13·87	0·968	23·8
85·7	14·3	—	11,340	278·3	13·9	0·918	24·3
79·1	20·9	—	11,808	280·2	14·0	0·859	25·0

The engine was started with petrol. The temperature of the cooling water in the cylinder head was maintained at about 208° F.

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[1] *Automotor*, Vol. IX, p. 414.

During the German trials of alcohol motors in 1902 a horizontal Deutz engine of 8·3 ins. bore and 11·8 ins. stroke, with a volume ratio of compression of 8·9, using a fuel composed of 86·1 per cent. by weight of alcohol and 13·9 per cent. of water, having a heat value stated as 9893 B.Th.U. per lb., consumed at the maximum of 16·55 BHP only 0·813 lb. of fuel per BHP hour, corresponding to the very high brake thermal efficiency of 31·6 per cent.

CHAPTER VII

PETROL ENGINES

BARNETT, in 1838, states in his Gas Engine Patent Specification that the engine could also be worked by some easily volatilised hydrocarbon, showing that even then the idea of using such light liquid fuels existed.

In 1867 the atmospheric gas engines of Otto and Langen were sometimes worked with carburetted air, obtained by passing it over the surface of what is now termed petrol.

Probably the first engine worked direct with petrol was that of J. Hock, of Vienna, in 1873; these stationary engines were afterwards built in Germany by the Maschinenfabrik Humboldt at Kalk.

The Hock engine operated similarly to the Lenoir, petrol vapour and air being employed instead of coal gas and air.

The Brayton engine (Vol. I, pp. 20–25) also used petrol, or 'light petroleum,' as it was then termed, in 1876.

Another early stationary engine using petrol was that of Messrs. Wittig & Hees, about 1880.

The change from the larger slow-running internal combustion engine to the small high-speed light liquid fuel motor, though in theory simple, proved to be practically a lengthy and difficult task. To the skill and perseverance of Gottlieb Daimler (1834–1900) is largely due the credit of this successful adaptation of the Otto cycle gas engine, whereby modern automobilism has become a practical achievement.

Up to about 1883 even the smallest internal combustion engines did not exceed a speed of roundly 200 revolutions per minute; Daimler's engine of that date ran at upwards of 800 revolutions; his design was of the inverted-vertical type, with 'splash' lubrication and enclosed crank-chamber. He first used the handle-starter. The valves were in a pocket on one side of the cylinder, the automatic inlet being placed above the exhaust; between them was fitted the open hot tube ignitor. Governing was effected by raising the exhaust valve; this was not, however, due to Daimler, Messrs. Koerting Bros. having already used it for some years previously.

Daimler used a surface carburettor, the air passing through a constant thickness of liquid petrol; the air was warmed before admission to the carburettor. The general arrangement of his motor is illustrated in Chap. III, fig. 204, of this volume.

His principal achievement was the successful employment of high speeds of rotation, which at once enabled the bulk and weight of the engine to be largely reduced without any sacrifice of power.

Daimler's first motor bicycle ran in 1886; the first car fitted with a Daimler motor ran in 1887. Later he turned his attention to the propulsion of launches and canal-boats by means of his motors, which were widely used in this way from 1887 onwards.

In 1889 Messrs. Panhard & Levassor concluded arrangements for the manufacture of the Daimler motor in France.

Contemporaneously with Daimler, Benz at Mannheim devoted much attention to the production of an internal combustion motor suitable for use in self-propelled road vehicles. His first design was a horizontal engine which was practically a small scale copy of the larger stationary type built by his firm, and this was retained up to about 1900.

The old Benz belt-driven cars with this horizontal engine at the rear were built in large numbers, and earned a high reputation for trustworthiness. Up to 1898 most of the German and Italian cars were fitted with the Benz motor; many were also run in France and England.

M. Charles Benz constructed his first small motor in 1878; this was of ¾ HP, with electric ignition; it was fitted to a tricycle, and a speed of 7 miles per hour was attained.

In 1888 the Benz motors and vehicles were introduced into France by M. Roger, who showed one at the Paris Exhibition of 1889; this was the sole representative of automobilism on view there.

D. Clerk examined this very early vehicle, and the following description by him is of some historical interest in the light of the immense development that has since taken place; writing from Paris, he said:

'There is a locomotive cab driven by a horizontal petroleum[1] engine constructed by E. Roger, of Batignolles, Paris. It has three wheels, like a Bath chair; the front wheel is used for steering, and the other two wheels are keyed on an axle connected to the engine by a pitch chain and bevel wheels. The petroleum engine is of the usual Otto type, and the flywheel lies in a *horizontal* plane, i.e. the crankshaft is vertical, while the engine is horizontal. A battery under the seat supplies current to ignite the explosive mixture, and the petroleum reservoir is conveniently stored away in the same place. Bevel wheels are driven by a belt from the engine, and a fast and loose pulley and belt shifter allow the connection to be interrupted without stopping the engine; the whole contrivance looks crude and complex.'

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[1] Really 'light petroleum,' i.e. petrol.

This account appeared in 'The Practical Engineer' for July 26, 1889.

In 1898 Messrs. Benz marketed three types, viz.:

(1) A single-cylinder, horizontal, Otto cycle, with electric ignition;

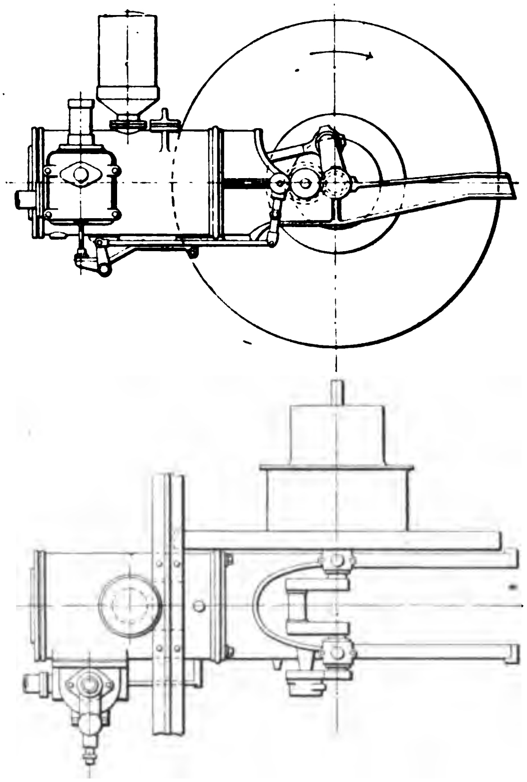


FIG. 285

(2) a two-cylinder ditto, the two cylinders forming one casting; and

(3) for the heavier classes of work, a two-cylinder horizontal, the cylinders being *vis-à-vis*.

Figs. 285, 286, and 287 illustrate these in outline. The ignition was by coil and battery; the sparking-plugs used had platinum wire electrodes, porcelain insulated.

Messrs. de Dion, Bouton & Co. have taken a prominent position in perfecting the petrol engine for use on motor vehicles since about 1895.

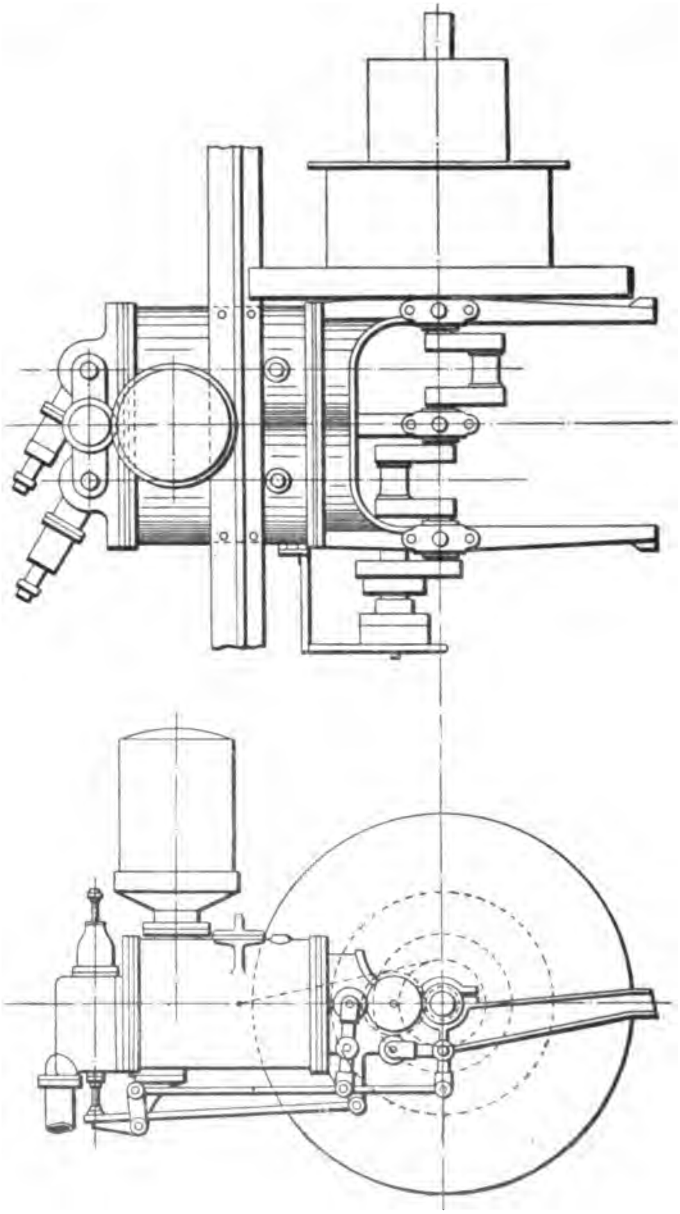


FIG. 286

In their hands the speed was soon successfully raised to 1500 or more revolutions per minute.

M. de Dion originally constructed a steam-driven tricycle, but later devoted his attention to the petrol motor; in order to reduce weight he ran his motors very fast, and by skilful design thus obtained a remarkably high output from his little engines. Thus one of his early single-cylinder motors, 2½ ins. × 2¾ ins., developed 1¾ BHP. High-tension ignition was fitted, a plug, non-trembling coil, and

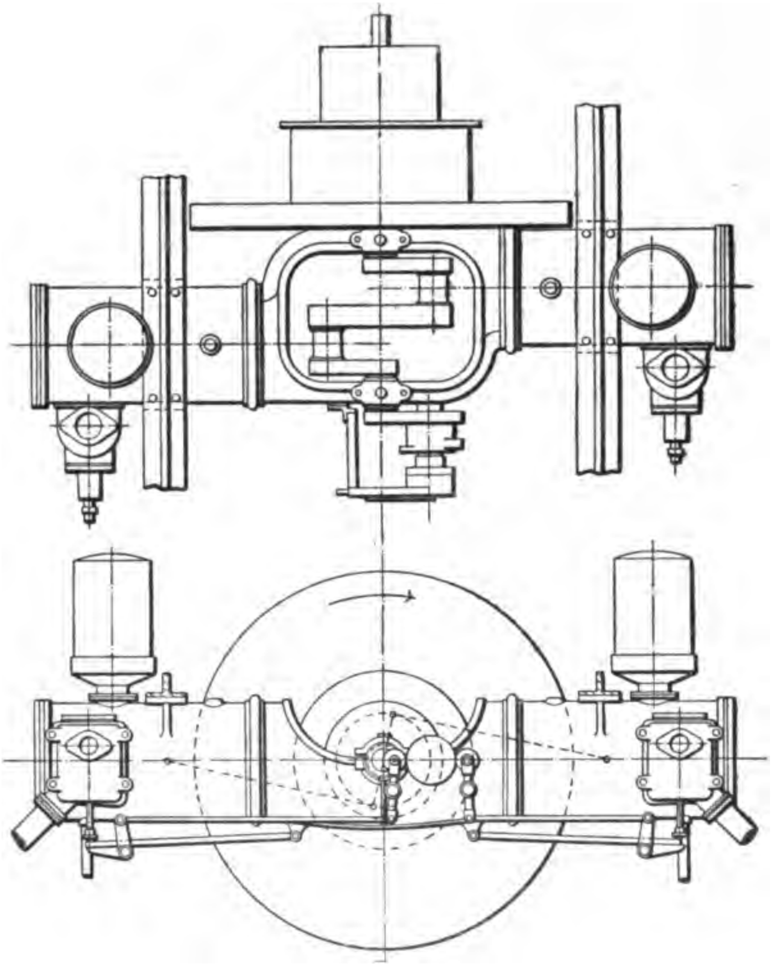


FIG. 287

dry battery being employed. The control was by hand on the throttle.

Later, engines for tricycles were constructed of higher power, as much as 14 HP being used on some racing machines; those fitted with engines of from 1¾ to 2¾ HP were, however, the most successful from the point of view of the ordinary user.

Messrs. de Dion were among the first constructors to employ aluminium for the crank-chambers of their engines.

Their original motor was air-cooled, having the now familiar fins cast with the cylinder. The flywheels were contained within an enclosed aluminium crank-chamber; lubrication was by the 'splash' method. The valves were in a pocket on one side of the combustion chamber, the automatic inlet being placed above the exhaust; the sparking-plug was placed between them. The revolution-speed was 1500–1800 per minute.

The early heavier type cars of Daimler and Panhard were fitted with governed engines usually having two cylinders about 3½ ins. × 5 ins., and running at about 700 revolutions per minute, with tube ignition. Later the two-cylinder engine was duplicated, and a four-cylinder engine thus produced. Messrs. Panhard also built some three-cylinder engines, a very perfect balance being thus obtainable; this design, however, has not survived. The lighter cars of the Argyll Co., Renault Frères, the New Orleans Co., &c., used small, throttle-controlled, electrically-fired, air-cooled, fast-running engines of the type with which the de Dion Co. were identified; over-heating troubles soon caused the adoption of water-cooling to become general. Water-cooled high-speed engines of much increased power were soon produced by de Dion, the Aster Co., Simms, and others.

Four-cylindered engines began to become general after 1902; in the Paris Automobile Exhibition of that year Messrs. Rochet-Schneider had a four-cylinder, 12 HP engine. Messrs. Buchet showed a four-cylinder 40 HP engine of 110 × 120 mm. (4·33 ins. × 4·73 ins.), designed to run at 1800 revolutions per min.; this was the type of motor used by M. Santos Dumont in his flights around the Eiffel Tower. The Daimler Co. also produced a standard four-cylinder design at that date, and there were several others.

Early estimates of the power required to propel motor vehicles seem to us now ludicrously inadequate. A correspondent writing in November 1896, with reference to a Benz car of that epoch, observes:

'It is found that a car of 1½ HP is not quite equal to carrying two people up the Yorkshire hills in bad weather, but an addition of ½ HP would probably be ample.'

About the same date Sir D. Salomons, Bt., in a public address, stated that the HP found to be necessary for cars had been raised from 3 or 4 to 6. He expressed the opinion also, that anything under 8 HP would not be found of much service if an average speed of 12 miles per hour were required.

Yet within a very few years of this, cars were running fitted with engines of 60, 80, and even 100 HP; since 1909, however, a combination of circumstances has resulted in the engine power of cars being reduced to a range of between 15 and 35 HP for all ordinary cases.

ENGINE WEIGHT PER HORSE-POWER

Statements of weight per HP, to be explicit, should clearly indicate whether the flywheel and immediate engine accessories (e.g. radiator, pump, silencer, carburettor, magneto, &c.) have been included. Motors for aeroplanes appear to be very light, due in part to the absence of a flywheel and to the fact that many of the types are air-cooled.

The first stationary petrol engines, used about 1880, weighed about 1100 lbs. per HP. Only six years later, in Daimler's early motors, this was reduced to 88 lbs. With increase in speed and improvements in designs and materials, the figure was rapidly reduced further, and the de Dion air-cooled bicycle motor of 1896 weighed only 26 lbs. per HP developed. In the car engines of 1911 the range, including flywheel, was from about 18 to 24 lbs. per normal BHP. Mr. F. W. Lanchester does not include the flywheel, regarding its size as to some extent a matter of taste on the part of the designer; if we exclude the flywheel, the weight per normal BHP of sustained load varies from about 8 to 16 lbs. only at the present time.

In most aeroplane engines so far constructed extraordinary pains have been taken in order to save weight, and in many cases these engines have but a short working life as a consequence. The average weight per BHP of the Gnome, Antoinette, and Renault aeroplane engines is only about 5½ lbs. In the seven-cylinder Gnome the weight is reduced to about 3¾ lbs. only; even lower figures than this have been stated, but these appear to be based upon estimated, and not upon actual powers (*vide* Critchley, 'Motors for Aerial Navigation,' 1910).

Where prolonged periods at full load working are necessary, fuel economy and durability are essential, even at the expense of increased engine weight. Fig. 288 is a diagram from Mr. Lanchester's Report of June 3, 1910, to the Aeronautical Committee, showing the influence of the continuous working range in hours on the total weight per BHP of the engine and fuel, the weight of flywheel and silencer being excluded.

It will be noted that in his view the very small weights of aeroplane engines at present seriously limit their range of sustained effort; for periods exceeding about 10 hours continuous full load running, the weight per BHP of the aeroplane motor becomes of the same order as that of a well-designed car engine.

In the construction of aeroplane engines nickel steel enters very largely, and in some designs (e.g. the Gnome) no aluminium whatever is employed, the weight reduction being obtained by a judicious employment of the former material only.

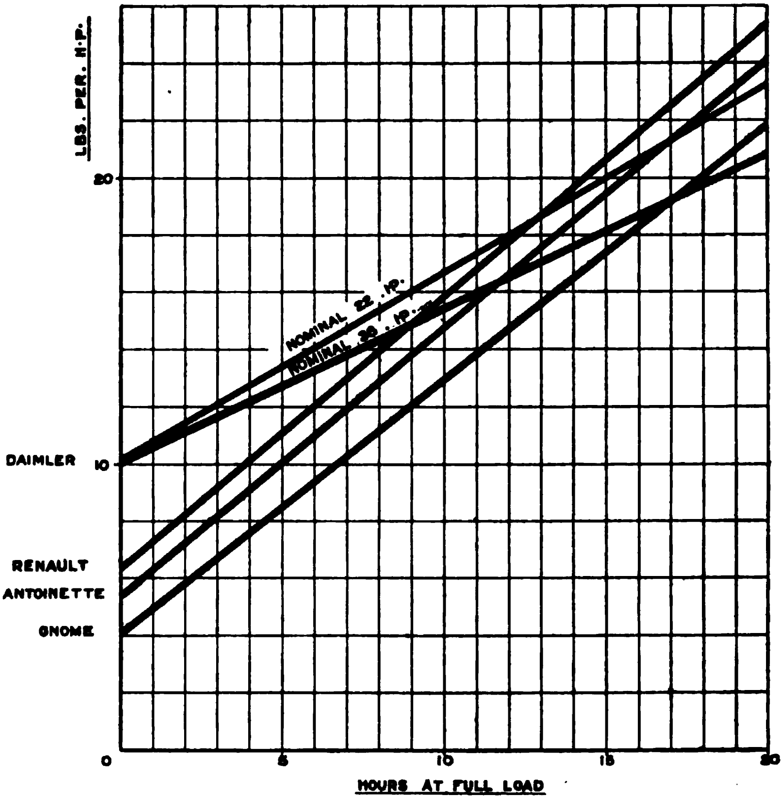


FIG. 288

The engine of a motor car is rarely called upon to develop its maximum power in ordinary usage, and then only for very short periods; in this respect the aeroplane engine is at a disadvantage, as it must be continually running at, or near, full power. Hence, to ensure durability, it seems probable that these engines will be found more substantially built in future; too much value has perhaps been set upon weight saving up to the present.

Petrol (and paraffin) engines used in marine propulsion are called upon to run for very long periods at full load, and in these it will be noted that the construction, to ensure durability, is considerably more massive than in the engines of cars. For marine motors, excluding racing boats, a piston speed of about 750 ft. per minute is rarely exceeded; the weight per BHP is consequently considerably greater than in car engines, in which piston speeds of 1200 feet per minute, or even more, are not unusual.

TYPES OF ENGINES

The inverted-vertical type is all but universal at present for car and boat work; for aeroplanes, vertical, diagonal, horizontal, radial, and rotary engines are employed; experience has not yet produced uniformity.

Of the inverted-vertical type as used in cars, the single- and two-cylindered engines are now few, and found only in the smaller and lower priced vehicles; four cylinders greatly preponderate, while the six-cylinder engine has been somewhat more largely used during 1911 than before. Of the eight-cylinder type the 35 HP de Dion is practically the sole representative.

In 1911 about 550 different designs of motor vehicles propelled by petrol engines were offered on the British market; the following is an analysis showing the numbers of 1, 2, 4, 6, and 8-cylinder engines used:

Single cylinder	15
Two cylinders	48
Four cylinders	414
Six cylinders	79
Eight cylinders	1
Total	557

In the case of four-cylinder engines some makers fit four separate cylinders cast from the same pattern, especially in the larger sizes. The cylinders are then identical in all respects, a faulty cylinder is easily replaced at small cost, and the separate cylinders are light and easily removable for examination of the piston, gudgeon pins, barrel, and combustion chamber; there is also no distorting tendency between the several cylinders. An arrangement more commonly adopted is to cast the cylinders in pairs. The effort to simplify the engine of a car in external appearance has probably been a factor in deciding many makers to adopt the practice of casting the four cylinders in one piece, or 'en bloc,' as it is termed. This gives a neat and compact engine in small sizes, but is awkward and heavy to handle in the larger types. In some cases a portion of the inlet, and of the exhaust passage also, is included in the cylinder casting; this does not, however, appear to be good practice, as heating of the incoming charge should be avoided, while, on the other hand, the exhaust gases should be led away from the cylinders as directly as possible to facilitate engine cooling. The 'en bloc' or 'monobloc' arrangement lends itself conveniently to thermo-syphon cooling, now becoming general.

After long service the cylinders of petrol engines frequently fail by breaking away at the junction with the flange or lugs by which connection is made with the crank-chamber casting. Failures also occur by development of cracks in the valve pockets, jackets, and working barrel. Thus the practice of employing separate cylinders is advantageous from this point of view. On the other hand the engine is heavier, and longer, with separate cylinders; moreover, separate cooling-water exits and inlets are needed for each cylinder, and also in many cases jacket covers are fitted to each cylinder, which increases the cost of construction. The method of using cylinders in pairs gives a shorter engine, and is well adapted to three-bearing crankshafts, as the two cylinders forming a pair can be kept close together, while between the two pairs there is room for an adequate central crankshaft bearing, fig. 289. With the cylinders cast 'en bloc' a short, stiff engine is produced, and there is a saving of water connections and jacket covers. Thus the cylinder arrangement appears to be largely a commercial question.

The number of crankshaft bearings is not yet settled. Many makers use five, as Lanchester, White & Poppe, Austin, Crossley, and Maudslay, and thus very rigidly support the shaft; in many other cases, as e.g. the Talbot, Wolseley, Germain, Renault, de Dion, Metallurgique, Adler, &c., three only are provided, the cylinders of each pair being kept as close together as possible. In the four-cylinder, 10 HP de Dion, 14 HP Delage, 14 HP Argyll, and a few other cases the crankshaft is borne by *two* bearings only; the cylinders are cast 'en bloc' and kept as close together as possible; an illustration of the de Dion crankshaft is shown in fig. 296. It is impossible to say that either the three- or five-bearing arrangement is the better; the three-bearing design is simpler and lighter, and it is at least certain that the very many three-bearing touring-car engines work with complete success. In these the crankshaft is of somewhat larger diameter than with the five-bearing arrangement, in order to provide the necessary increased stiffness.

With six cylinders also, practice is not yet uniform. The latest six-cylinder, 38 HP Lanchester engine, for example, has the cylinders cast separately; this is also Messrs. White & Poppe's practice. On the other hand, in the 24 HP, six-cylinder 'La Buire' engine, the cylinders are 'en bloc' (fig. 290). Between these extremes we have the cases of three pairs, as in the Napier, and the practice, apparently initiated by Mr. Royce, of arranging the cylinders in two sets, each of three; this latter arrangement is growing in favour, and has been adopted by several other makers, e.g. the Sunbeam Co., the Vauxhall Co., Delaunay-Belleville, &c. It forms a convenient disposition, and gives a good bearing midway along the crankshaft, without unduly increasing the overall length of the engine.

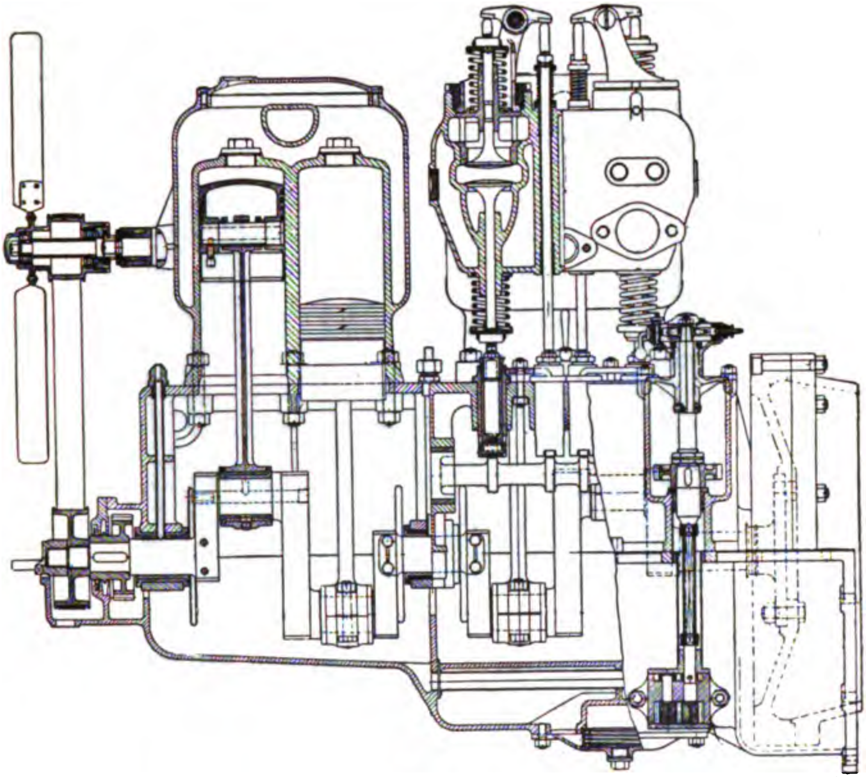


FIG. 289

For all ordinary purposes sufficiently good balance and evenness of torque can be obtained with a four-cylinder engine. Where unusual steadiness of torque is desired the six-cylinder engine may be employed (*v.* fig. 242).

The torque of the eight-cylinder type is still more nearly constant, but its balance is not so good as that of the six-cylinder engines. In engines with fewer than four cylinders the uniformity of torque diminishes with the number of cylinders. So far as balance goes, the three-cylinder engine may be made as satisfactory as the four-cylinder, but its torque is necessarily less uniform (*v.* Chap. IV, fig. 242). A few three-cylinder car engines were built some years ago by Messrs. Panhard & Levassor, the Vauxhall Co., and others, but the type has not survived. The relative advantages of four and six cylinders have been much discussed; undoubtedly increased smoothness of running can be obtained from the six-cylinder type, but some difficulty has been experienced in providing sufficient crankshaft stiffness in these somewhat lengthy engines.

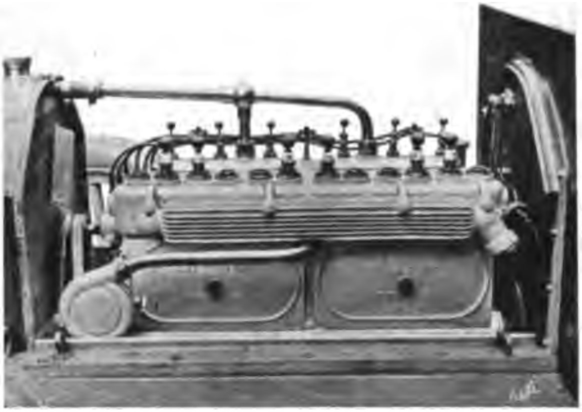


FIG. 290

In the 1911 six-cylinder, 18–22 HP engine of the Sunbeam Co., the cylinders, as already stated, are in two sets, each of three, and the crankshaft is supported in seven bearings, the first, fourth, and seventh of which are considerably longer than the remaining four. The crank throws are shown in fig. 291, the arrangement giving the equivalent of two three-cylinder engines in tandem, with opposed rocking moments; the balance is accordingly very perfect. This arrangement of six-throw crankshaft is now very generally adopted.

The crank disposition is also clearly shown in the end view diagram; the first and sixth throws are together, as also the second and fifth, and the third and fourth.

In fig. 292 the six-cylinder 24–30 HP engine of the Wolseley Co. is shown in section; the cylinders are here cast in pairs, the members of each pair being kept as close together as possible. Here also the crank throws give the opposed three-cylinder advantage; Nos. 1 and 6 are together, 2 and 5, and 3 and 4.

It will be noted that the crankshaft in this design is borne by four bearings only.

Another arrangement occasionally adopted is that shown in fig. 293, which also illustrates a four-bearing, six-throw shaft. Here the first and second cranks are opposed, as also the third and fourth, and the fifth and sixth. A good balance is thus obtained, the rocking moment of adjacent cranks being small, as they are close together; this disposition favours the grouping of cylinders in three pairs. The arrangement is the equivalent of three tandem two-cylinder engines with cranks at 180°.

A third arrangement possible is to put adjacent cranks together (fig. 294), giving three tandem two-cylinder engines with cranks at 0°. This gives the three-cylinder type of balance; but on account of the considerable length of the engine the rocking moments are considerable; accordingly this arrangement cannot be considered a good one.

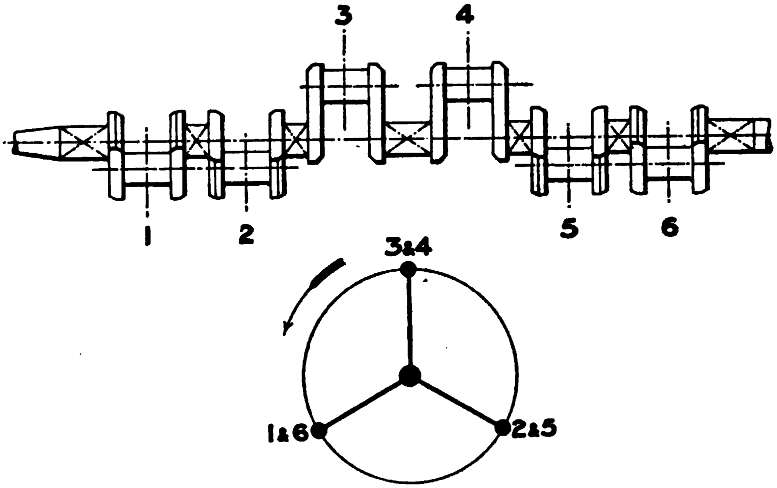


Fig. 291

In four-cylinder engines the four cranks are in one plane, the usual arrangement being to put the first and fourth together, as also the second and third. Fig. 295 illustrates the three- and five-bearing types in general use.

With this disposition a practically perfect balance is obtained.

When three bearings only are employed, it is usual to find the central one rather long; also in some designs of this type the crank pins are somewhat larger in diameter than the shaft; for example, in the 40 HP Wolseley engine, the shaft is 2⅛ ins. diameter, while the

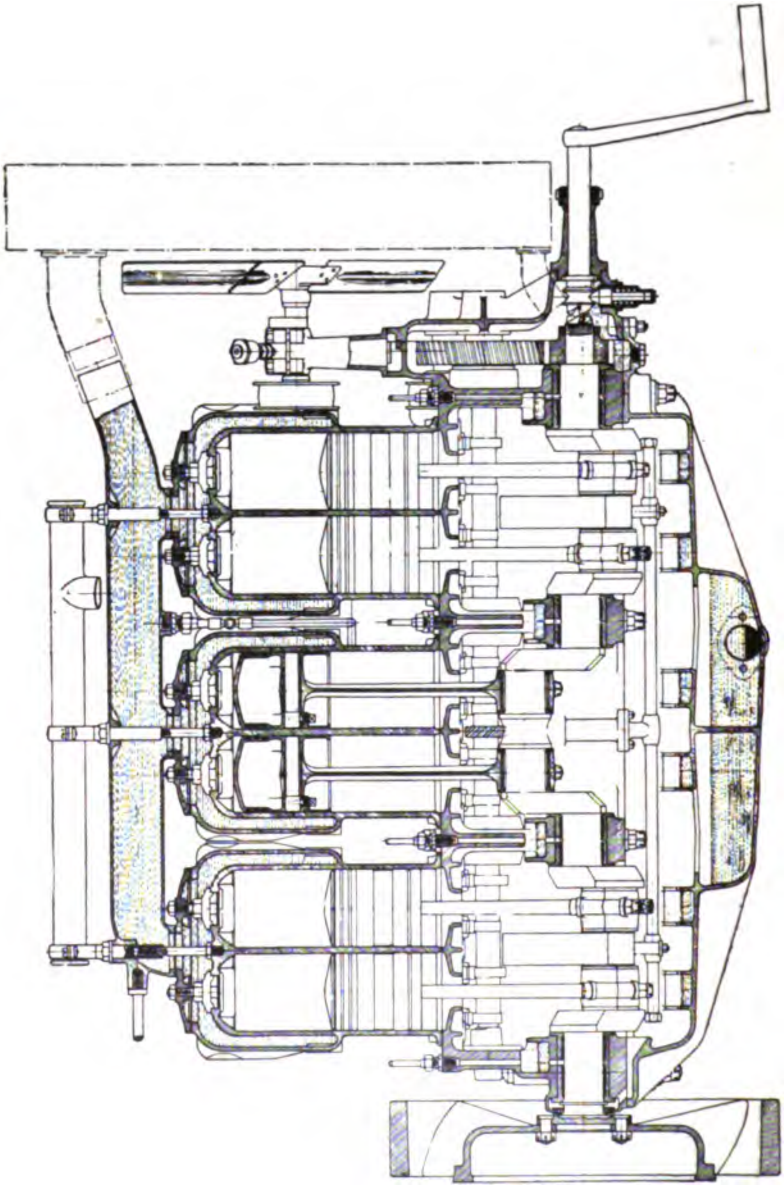


FIG. 292

pins are 2¼ ins. diameter; more usually, however, the crank-pins and shaft are equal in diameter.

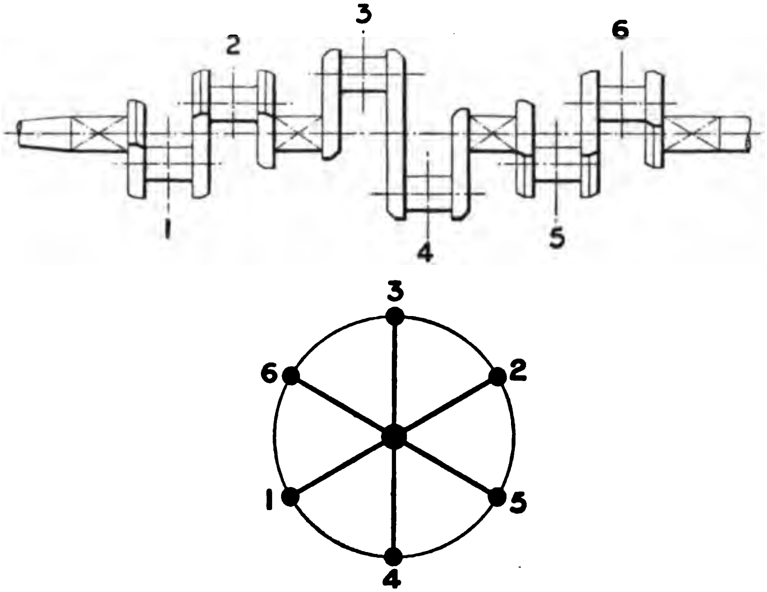


FIG. 293

Fig. 296 illustrates the unusual crankshaft arrangement adopted in the short four-cylinder 10 HP de Dion engine (2·60 ins. × 3·94 ins.),

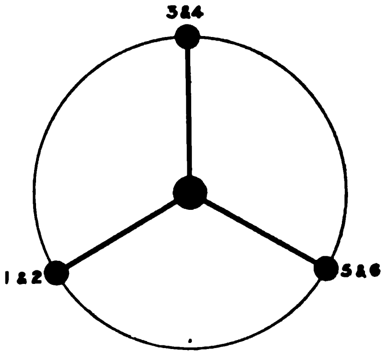


FIG. 294

and in the four-cylinder 14 HP Delage, and several others. The central bearing is here omitted altogether, and the crankshaft is supported in two end bearings only.

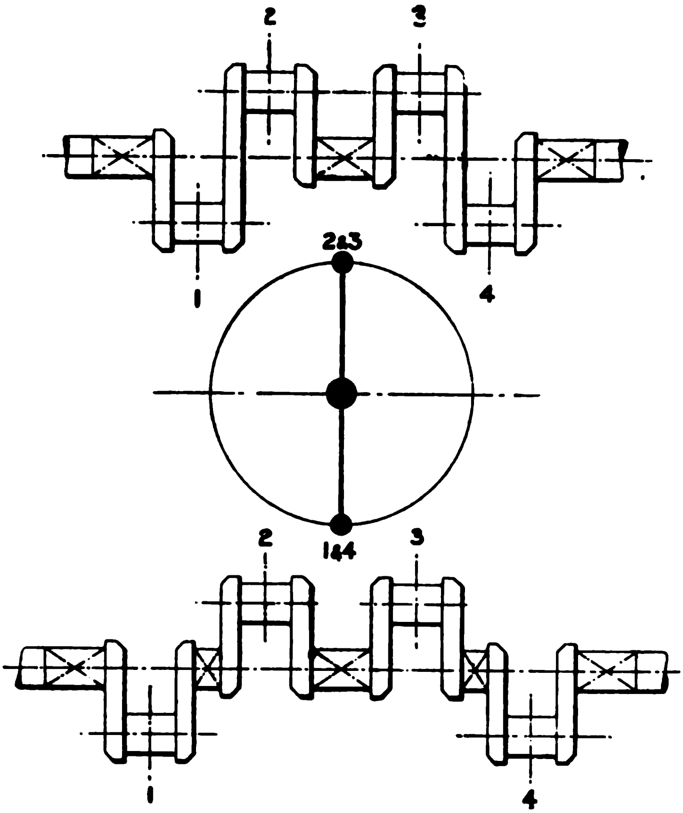


Fig. 295

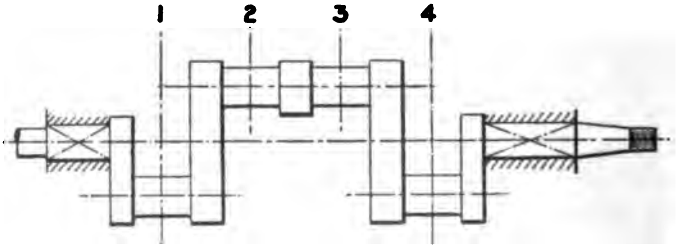


Fig. 296

Crankshafts of petrol engines are generally machined from steel stampings; in a few instances, however, built-up shafts are still used.

The material employed may be the hardest steel that can be satisfactorily worked. The Maudslay Co. have used a steel having an elastic limit of only 27½ tons per sq. in., with an ultimate strength of 41 tons per sq. in., and 28½ per cent. elongation in a two-inch length (including the fracture) of a test specimen. The Wolseley Co. use Vickers' 'crankshaft and axle steel.' Messrs. White & Poppe employ a nickel steel. The Daimler Co. use a vanadium chrome steel. Messrs. Lanchester prefer a 3 per cent. carbon steel having an elastic limit in tension of 50 tons per sq. in., and an ultimate strength of 62 tons per sq. in. Suitable steels for the more important parts of car engines are given in the accompanying table. The crankshaft diameter is determined from considerations of *stiffness*, and not of strength alone, rigidity in working being of the utmost importance. For four-cylinder engines the average shaft diameter is about ⅓ of the cylinder bore for five bearings, and ½ the bore in the case of a three-bearing design. Where two bearings only are used the crankshaft diameter may be as much as ¾ of the cylinder bore. In some cases the crankshaft stampings are ground to the finished condition, no turning whatever being necessary.

SOME STEELS SUITABLE FOR MOTOR CAR ENGINES AND OTHER PARTS

Part of chassis	Tensile tests				Suitable steel alloy
	Elastic limit, tons per sq. in.	Ratio of elastic to maximum, per cent.	Maximum stress, tons per sq. in.	Elongation, per cent. in 2"	
Crankshafts .	48·5	86·5	56·0	22·0	Chrome nickel
,, .	48·0	85·7	56·0	19·0	Chrome vanadium
Connecting-rods (stamping) .	30·0	62·5	48·0	26·0	5 per cent. nickel
Gear Shafts (*a*)	98·0	89·0	110·0	13·0	Chrome nickel
,, ,, . (*b*)	88·0	78·5	112·0	2·5	
Live axles .	57·0	77·0	74·0	18·0	Chrome nickel
	48·0	65·7	73·0	14·0	
Piston gudgeon pins	29·0	85·2	34·0	13·0	Ubas annealed case-hardening steel.

BEARINGS

The crankshaft and big-end bearings are usually of white metal; the gudgeon bearing is usually of bronze.

Phosphor bronze bearings and case hardened shafts have been used, and form an exceedingly durable combination, which may again come into favour when crankshafts are relieved from unnecessary bending actions induced by the warping of the engine frame and imperfections in the mode of connection between engine and gear-box shafts.

Any twisting of the engine frame or bending of the shaft tends to cause intense local pressures in the main bearings whereby the oil film may be crushed, with consequent rapid wear of the rubbing surfaces. Three-point suspension of the engine, and designs in which the two universal joints in the intermediate shaft between engine and gear-box, when fitted, are kept as close as possible to the gear-box and engine respectively, are engaging much attention. Designers of experience maintain that by the elimination of these unnecessary straining actions the life of the bearings may be increased by fully 25 per cent.

FIRING ORDER

Almost all petrol car engines operate on the Otto cycle; accordingly, in the single-cylinder type there is one working impulse only in every two revolutions of the crank-pin, fig. 297.

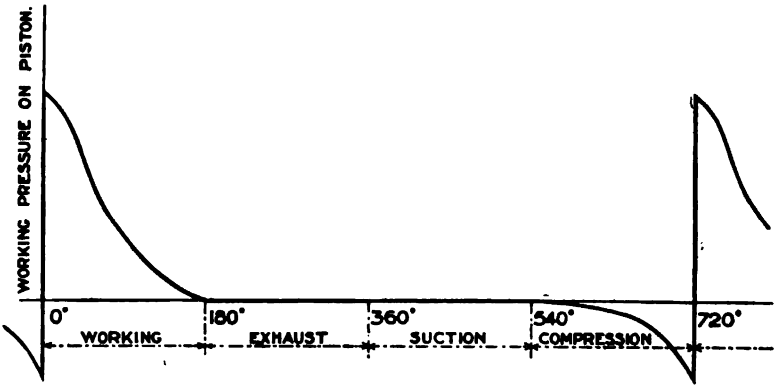


FIG. 297

With two cylinders three arrangements are practicable, viz.:

(A) The crank-pins may be together (i.e. at 0°) and both cylinders on the same side of the crankshaft.

(B) The crank-pins may be opposite (i.e. at 180°) and both cylinders on the same side of the crankshaft.

(C) The crank-pins may be at 180°, and the cylinders on opposite sides of the crankshaft.

(A) With the crank-pins together there is a working impulse in every revolution, as indicated in fig. 298; the balance, however, is bad, as both pistons ascend and descend together, and this type has accordingly not been much used. Now that reciprocating parts can be made so very light, it is possible that quite satisfactory engines may be thus built, and so secure the advantage of an impulse every revolution; this method has, in fact, been recently adopted in the two-cylinder 9 HP de Dion engine of bore 2·95 ins. and stroke 5·12 ins.

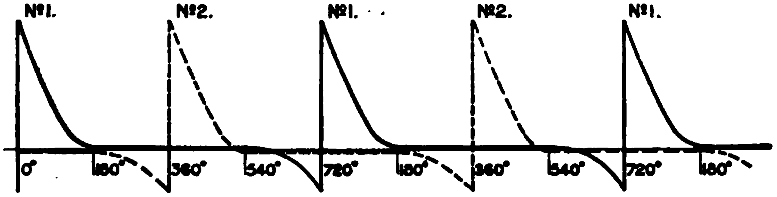


FIG. 298

(B) This is the usual arrangement of the two-cylinder engine; with the crank-pins at 180° the reciprocating parts are nearly balanced; the balance would be perfect but for the effect of the connecting-rod obliquity. On the other hand, the impulses are much more irregular than in type (A), as the two working strokes necessarily follow one another in the same revolution, leaving the complete second revolution idle; the distribution of the impulses is shown in fig. 299, and may be compared with fig. 298, illustrating the A-type arrangement. The torque is much less uniform than in the previous case.

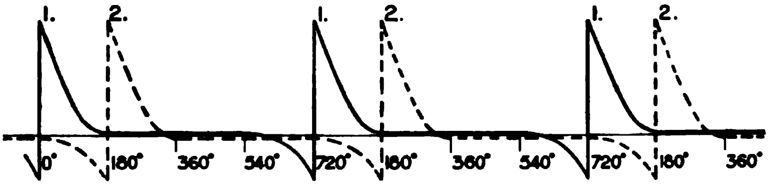


FIG. 299

(C) When the cranks are at 180° and the cylinders are on opposite sides of the crankshaft, there is perfect balance of the reciprocating parts, and also a working impulse every revolution. But this arrangement practically necessitates a horizontal engine, which is not usually considered convenient for car work, and the type is also somewhat bulkier, heavier, and more expensive to build than either (A) or (B); it has accordingly been rarely used. A recent very successful application of this type is found, however, in the engine of the 'Douglas' motor bicycle.

THREE-CYLINDER ENGINES

The cranks are here at 120°, and the accompanying fig. 300 illustrates roughly, and apart from inertia effects, the distribution of the working impulses along the crank-pin path. As each cylinder gives one impulse in two revolutions, three cylinders furnish three such impulses, so that with a three-cylinder engine we get 1½ impulses per crankshaft revolution. In the position indicated by the crank-diagram in the figure, 1 is commencing its working stroke; 2 is concluding its suction stroke; while 3 is performing its exhaust stroke.

The order of firing must obviously be either 123, 123 . . . or 321, 321 . . .

FOUR-CYLINDER ENGINES

The normal four-cylinder engine, with crankshaft arrangement as in fig. 295 (*ante*), permits of two different orders of firing; for if No. 1 fires during the first half revolution of the crankshaft, either No. 2 or No. 3 *must* fire during the second; No. 4 must in either case fire during the third; and finally either No. 3 or No. 2 will fire during the fourth.

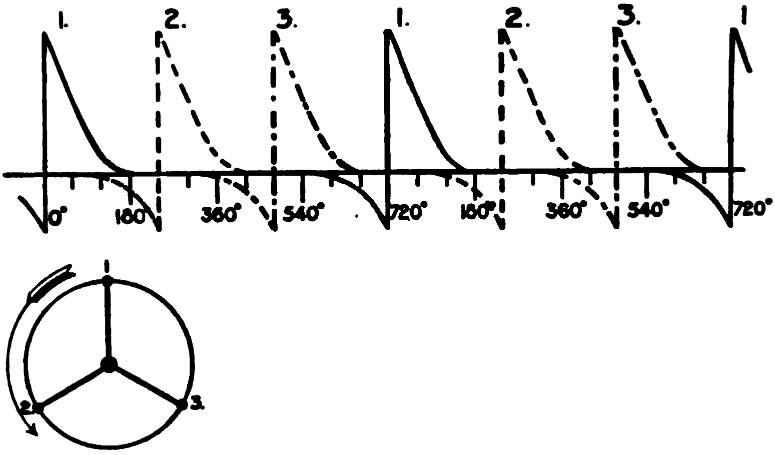


FIG. 300

Hence the two firing orders are:

1243, 1243 . . . and 1342, 1342 . . .

Both are employed, but 1342 appears to be most usually adopted.

Fig. 301 shows approximately the distribution of impulses along the crank-pin path; there are here two working impulses in each revolution of the crankshaft, and consequently the torque is more nearly uniform than in the preceding cases.

SIX-CYLINDER ENGINES

Here there are six impulses during every two revolutions of the crankshaft, or three per revolution; the torque is accordingly more nearly constant than in the previous case; the distribution of the impulses is shown in fig. 302.

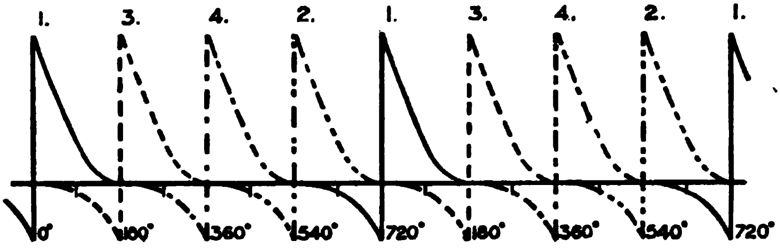


FIG. 301

As to possible orders of firing with the two three-cylinder opposed disposition of cranks usually adopted, it may be noted that there may be two crank arrangements, viz. as indicated at A and B in fig. 303, and that B is a mirror image of A.

Referring to A, the engine may be regarded as composed of the following four three-cylinder pairs:

1 2 3 and 6 5 4
1 2 4 and 6 5 3
1 5 3 and 6 2 4
1 5 4 and 6 2 3

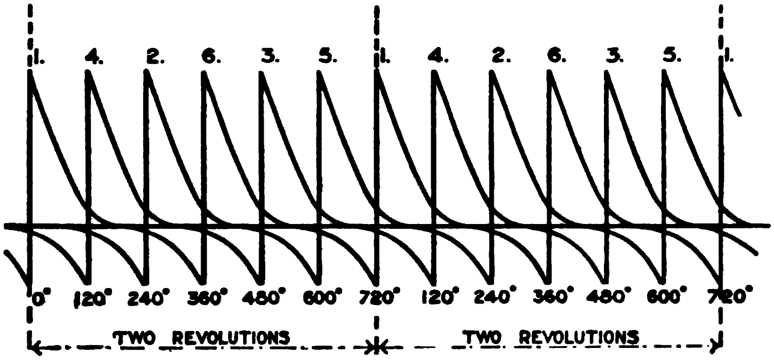
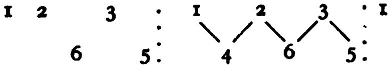


FIG. 302

Now 123 fire in the order stated, the crank angles described between the firing instants of 1 and 2, and of 2 and 3, being each 240° (*v.* fig. 300).

But cranks 1 and 6 being together, 6 must necessarily fire when 360° of crank angle has been described from the firing point of 1.

Hence 6 must fire between 2 and 3. Similarly for 2 and 5, and for 3 and 4. Thus we get the following firing order:

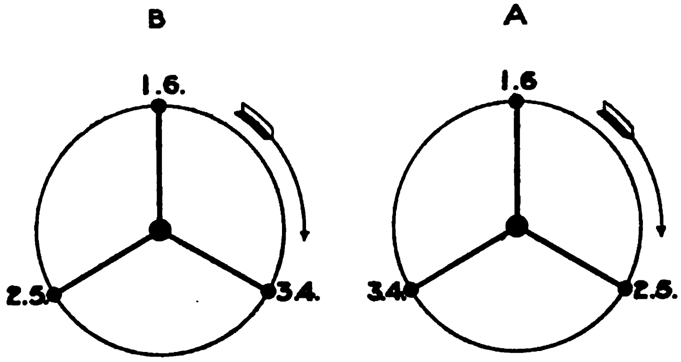


Similar considerations apply to the remaining three pairs; hence we have, on the whole, four possible firing orders with the A disposition of cranks, viz.:

142635, 132645, 145632, and 135642.

Similarly, for the B case a further four orders are possible, as shown in fig. 303; thus on the whole there are eight firing orders that may be used.

Messrs. White & Poppe adopt the sequence 142635, as with this order no two cylinders interfere during suction and exhausting; the Wolseley Co. also use this sequence. Messrs. Lanchester at first used 236541, but found that in their design a quieter and steadier engine resulted from replacing this by the sequence 153624, which is the image of the White & Poppe order. These are the only two orders wherein two adjacent cylinders do not fire consecutively, and are those in general use at the present time.



Orders of Firing

536241	142635
236541	145632
546231	132645
246531	135642

Fig. 303

Range of Size.—For car work the four-cylinder petrol engine ranges in size from about 2½ ins. bore × 3½ ins. stroke in the small 10 HP Austin, to 5⅛ins. bore × 7$\frac{1}{16}$ ins. stroke in the 90 HP Mercedes.

In six-cylinder engines the range is from the 15¾ HP, 65 × 125 mm. (2·56 ins. × 4·92 ins.) Delage, to the 6⅛ ins. × 5 ins. of the 90 HP

Napier. The most frequent size of car engine at present (1911) is about 80 mm. bore by 120 mm. stroke, i.e. about 3⅛ ins. × 4¾ ins.

Within recent years the ratio of stroke to bore has shown a tendency to increase. Few examples, however, exist at present in which the ratio has as high a value as 2; broadly speaking, a long-stroke engine, other things being equal, holds out better under up-hill 'collar work' than the short-stroke type; practice tends to the use of as large a ratio of stroke to bore in car engines as can be adopted without sacrifice of smoothness in running.

The long-stroke engine necessitates a heavier crankshaft and transmission gearing for given cylinder diameter, and thus, generally, a heavier chassis design, on account of the greater torque created. On the other hand the practicable maximum piston speed that can be regularly used increases with the value of the stroke-bore ratio. The higher values of the ratio are usually found in the smaller engines, the value diminishing in practice with increase in size of engine. The range is from 2·0 in engines of 6–15 HP, to 1·0 or even slightly less for 50 HP and upwards; the following short table from a list of car engines for 1911 illustrates broadly the state of current practice:

TABLE ILLUSTRATING BROADLY THE VARIATION IN RATIO OF $\frac{\text{STROKE}}{\text{BORE}}$; 1911 CAR ENGINES

Description.	Nominal HP	Bore, mm.	Stroke, mm.	$\frac{\text{Stroke}}{\text{Bore}}$	—
Gregoire	—	80	160	2·00	Small car engines
Lion-Peugeot	12·0	75	150	2·00	
Calthorpe	15·0	75	150	2·00	
Le Gui.	10·0	65	130	2·00	
Le Gui.	15·0	75	150	2·00	
Jackson	6·2	100	200	2·00	
Delage	15·7	65	125	1·92	
Opel	14·0	64	120	1·88	
Delage	9·5	62	110	1·78	
Sunbeam	18–22	80	120	1·50	Medium car engines
Talbot	20	80	120	1·50	
Sheff-Simplex	25	85	127	1·49	
Napier	30	82	127	1·55	
Deasy	18–24	90	130	1·45	
Crossley	20	102	140	1·37	
Dennis	40	126	130	1·03	Large car engines
Lanchester	38	102	102	1·00	
Maudslay	35–45	127	127	1·00	
Napier	65	127	127	1·00	
Talbot	35	127	120	0·945	
Imperia	56–60	150	140	0·933	
Napier	90	156	127	0·814	

The smallest value of the stroke-bore ratio is found in the 20 HP Lanchester engine, which has a bore of 4 ins. with a stroke of 3 ins.; the ratio is thus 0·75 only; in this respect this Lanchester engine is quite unique. A stroke of 3 ins. is also the smallest used in normal cars; the longest—excluding racing engines—is about 7¾ ins., though it is very rare to find an engine with a stroke exceeding 7 ins.

Pistons.—Pistons are usually of medium hard close-grained cast iron, generally with three, but occasionally four, cast-iron spring rings; an additional 'scraper' ring is sometimes fitted near the bottom of the piston to prevent oil from the crank-chamber passing up into the combustion chamber. In some cases the gudgeon pin is retained in position by a special ring in addition.

For normal car engines the piston is usually rather greater in length than the cylinder bore, the ratio being about 1·16 to 1 for the average case.

For the smaller sizes flat-topped pistons are usual. For medium and large sizes coned or flat-domed tops predominate; with pistons of 5 ins. diameter and upwards flat-topped designs have frequently caused trouble from over-heating. In a few cases, e.g. the Daimler, the piston crown is concave. The deposition of carbon is retarded by polishing the piston crown, and this practice is followed by many makers.

In a few cases cooling fins are cast on the under side of the crown to assist in the conduction of heat from the central highly heated portions of the surface; cooling of the piston is effected partly by heat conduction to the periphery and thence through the cylinder walls, and partly by the oil thrown from the crank cheeks upon the under side of the piston. In some cases the carburettor takes its air from the crank-chamber; this practice assists the general cooling of the interior of the engine, in addition to supplying warm air carrying some fine oil spray for the working mixture.

To allow for expansion, the diameter of the piston above the topmost ring is usually made about $\frac{3}{1000}$ of the diameter less than the cylinder bore; it has sometimes also been found necessary to ease away the metal of the piston in the neighbourhood of the ends of the gudgeon pin, owing to the distortion under the explosion pressure tending to jam the piston in the cylinder in the neighbourhood of these ends.

The spring rings are of cast iron, bored eccentrically, and split diagonally at the thinnest part; the ring exceeds the cylinder bore by about $\frac{1}{80}$, before it is sprung into place. When in place the split edges should be as close together as possible, in order to diminish leakage of gas through them ; if, however, they are allowed to approach too closely, the expansion when working may cause them to abut together, and the cylinder friction is then greatly increased ; engines have sometimes been found to pull up completely from this cause. Some designers fit 'snugs' in the pistons to prevent the spring rings turning round in working, so that all the split ends shall come into the same line ; others omit the snugs, but fit the spring rings with the split ends sloping alternately one way and the other. The Germain Co. have adopted double spring rings of an ingenious design as indicated in fig. 304 ; these are gas tight in all positions.

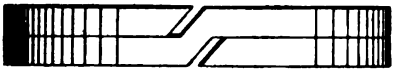


FIG. 304

A recent very light design of piston is shown in fig. 305, which illustrates the 'lantern' type adopted in the 3·15 ins. × 4·73 ins. Sunbeam engine. The material used is a special malleable cast iron, and the casting is machined inside and out.

There are three spring rings above the gudgeon, and a fourth 'scraper' ring at the bottom ; the trunk between gudgeon and scraper ring is not only very thin, but its sides are completely removed as shown in the sketch ; the lightening holes at the gudgeon level will also be noted. The design, although extremely light, is said to be of adequate strength.

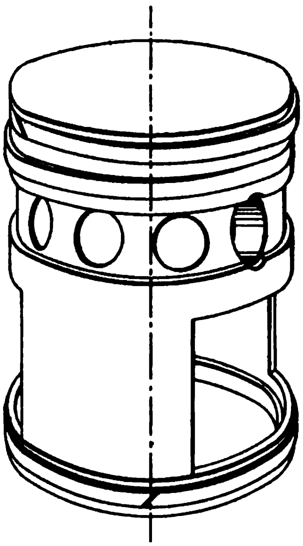


FIG. 305

The gudgeon pin is usually carried in bosses in the piston and fixed therein either by one or two steel set-screws, or by a special spring ring encircling the piston and ends of the pin. As usually designed it sustains a maximum load per sq. in. of projected area (i.e. bearing length × diameter) of from 2500 to 3000 lbs. per sq. in. ; the pins are occasionally hollow, but usually solid.

Reciprocating Parts.—It is a point of importance in these quick-running engines that the mass of the piston and connecting-rod be kept as small as possible in order to minimise vibration, friction, and inertia effects. The question of inertia is dealt with more fully later in this chapter; see also Chap. IV and Appendix I.

A few weights of reciprocating parts for 1908 engines are tabulated hereunder; further figures are given for engines of later design in the table on p. 505; it may be noted that, on the average, the piston and connecting-rod of these engines are, roughly, equal in mass.

MASS OF RECIPROCATING PARTS. 1908 ENGINES

Cylinder bore, inches	Stroke in inches	Piston mass complete, in lbs. cast-iron pistons.	Connecting-rod mass, lbs.	Total mass of reciprocating parts, lbs.
2·95	4·73	2·06	2·38	4·44
3·38	4·00	2·50	3·00	5·50
3·50	3·78	3·06	3·19	6·25
3·74	4·73	3·00	3·75	6·75
4·00	4·00	3·78	3·35	7·13
4·00	4·73	3·00	3·38	6·38
5·00	5·00	7·06	6·31	13·37
5·00	6·00	6·00	5·50	11·50

By using pressed steel pistons, machined all over, the mass may be reduced to about two-thirds of that with cast iron; such pistons, though more expensive to produce, are now being somewhat largely used; they appear to have been first introduced by Mr. Garrard in the 'Talbot' engines about 1907.

Connecting-rods are of stamped steel, usually of H-section; the ratio of length of rod to stroke of piston varies but little in modern practice from the value 2·25.

In the following table the joint weight of the reciprocating parts, i.e. piston and connecting-rod complete, is given for a number of standard car engines with cast-iron pistons.

If d denote the cylinder bore in inches, r the stroke-bore ratio, and m the mass of the piston and connecting-rod complete in lbs., then Burls (*Proc. Inst. A.E.*, Vol. V, p. 190) has proposed the empirical equation:

$$m = 0{\cdot}08\, d^3 (1 + 0{\cdot}15\, r) + 1{\cdot}5 \text{ lbs.} \qquad (1)$$

as satisfactorily resuming current practice; comparison of actual and calculated values of m can be made from the table hereunder. The range in cylinder bore is from less than 2½ ins. to 12 ins., and in stroke-bore ratio from 1·78 to 0·67. Equation (1) is of service in checking the mass of the reciprocating parts in new designs.

TABLE SHOWING ACTUAL AND CALCULATED MASS OF RECIPROCATING PARTS (CAST-IRON PISTONS) OF PETROL ENGINES

$$m = 0·08\, d^3\, (1 + 0·15\, r) + 1·5 \text{ lbs.}$$

1910 TYPE

Bore, d, inches	Stroke, s, inches	$r = \frac{s}{d}$	$d^2s =$	Actual mass in lbs.	Mass, m, by calculation
2·44	4·33	1·78	25·8	3·20	2·97
2·56	4·33	1·69	28·4	3·24	3·18
2·95	4·73	1·69	28·4	4·44	4·04
2·95	4·73	1·60	41·2	4·60	4·04
3·15	3·94	1·25	39·1	4·63	4·47
3·38	4·00	1·42	45·7	5·50	5·25
3·50	4·50	1·29	55·2	5·25	5·60
3·56	4·75	1·33	60·2	5·50	5·80
3·74	4·73	1·27	66·2	6·75	6·50
3·74	4·73	1·27	66·2	8·10	6·50
4·00	4·00	1·00	64·0	7·13	7·40
4·00	4·38	1·10	70·1	7·25	7·50
4·25	5·25	1·24	94·8	9·12	8·80
4·33	5·52	1·28	103·5	9·25	9·25
4·50	5·00	1·11	101·2	9·50	10·00
4·63	5·00	1·08	107·3	13·12	10·75
4·75	5·00	1·05	112·6	12·22	11·40
4·92	5·92	1·20	143·0	13·80	12·70
5·00	5·00	1·00	125·0	13·37	13·00
5·00	5·13	1·03	128·2	12·25	13·00
5·12	5·52	1·08	144·4	13·20	13·80
7·25	7·50	1·03	394·0	32·50	36·70
Marine Eng.					
8·00	8·00	1·00	512·0	54·0	48·60
12·00	8·00	0·67	1152·0	151·0	153·50

TABLE SHOWING ACTUAL AND CALCULATED MASS OF RECIPROCATING PARTS (PRESSED STEEL PISTONS) OF PETROL ENGINES, USING EQ. (2)

Bore, d, inches	Stroke, s, inches	$r = \frac{s}{d}$	d^2s	Actual mass, in lbs.	Mass calculated from (2)
3·13	4·5	1·44	44·1	3·72	3·36
3·13	4·75	1·52	46·5	3·43	3·38
3·15	4·73	1·50	46·9	3·53	3·41
3·38	4·25	1·26	48·6	3·38	3·8
3·35	5·0	1·49	56·1	3·5	3·8
3·75	4·5	1·2	63·2	3·63	4·6
3·75	4·5	1·2	63·2	3·88	4·6
3·94	4·73	1·2	73·5	5·3	5·1
3·94	5·12	1·3	79·3	5·52	5·15
3·94	6·30	1·6	97·8	4·91	5·28
4·0	5·5	1·38	88·0	4·87	5·12
4·0	7·0	1·75	112·0	5·5	5·53
5·0	5·25	1·05	131·0	10·5	8·73

For pressed steel pistons the appropriate equation for present designs (1910) is:

$$m = 0{\cdot}05\, d^3 (1 + 0{\cdot}15\, r) + 1{\cdot}5 \text{ lbs.} \tag{2}$$

and the short table on p. 505 shows the degree of closeness of calculated and actual values.

Messrs. White & Poppe express the mass of the reciprocating parts by a formula of the form $a\ (d + b)^3$, where a and b are constants; for inch-lb. units it is found that:

$$m = 0{\cdot}037\ (d + 1{\cdot}9)^3 \tag{1A}$$

for cast-iron pistons and steel rods very well resumes the facts over the practical range of car engine sizes.

The table hereunder enables comparison to be made between the figures furnished by the two formulæ (1) and (1A), and from this table the curves shown in fig. 306 are drawn. It will be noted that Eq. (1A) gives lower values for m than Eq. (1) as the size of the engine increases; reference to the table showing actual and calculated values of m shows, however, that Eq. (1) closely resumes the actual figures over the complete range of sizes given.

TABLE OF CALCULATED VALUES OF m FROM EQS. (1) AND (1A)

Bore in inches, d	Stroke-bore ratio, r	m in lbs. by calculation from	
		Eq. (1)	Eq. (1A)
2·5	1·8	3·10	3·15
3·0	1·8	4·24	4·35
3·5	1·6	5·75	5·82
4·0	1·5	7·77	7·58
4·5	1·4	10·3	9·68
5·0	1·2	13·3	12·1
5·5	1·1	17·0	15·0
6·0	1·0	21·4	18·2
6·5	1·0	26·7	21·9
7·0	1·0	33·1	26·1

In the following table values of m derived from Eq. (1), and of $\frac{d^2 s}{m}$ and its square root—which will be useful later—are exhibited for all cases likely to occur in car engine practice:

TABLE GIVING CALCULATED VALUES OF m, FROM EQ. (1), CAST-IRON PISTONS

$r =$	d in inches	$d^2s =$	$m =$ in lbs.	$\frac{d^2s}{m} =$	$\sqrt{\frac{d^2s}{m}} =$
0·75	4·0	48·0	7·2	6·67	2·58
	4·5	68·3	9·6	7·12	2·67
	5·0	93·8	12·6	7·44	2·73
	5·5	124·8	16·3	7·67	2·77
	6·0	162·0	20·7	7·83	2·80
1·0	3·0	27·0	4·0	6·75	2·60
	3·5	42·9	5·4	7·94	2·82
	4·0	64·0	7·4	8·65	2·94
	4·5	91·1	9·9	9·20	3·04
	5·0	125·0	13·0	9·63	3·10
	5·5	166·4	16·8	9·92	3·15
	6·0	216·0	21·4	10·1	3·18
1·25	3·0	33·8	4·1	8·24	2·87
	3·5	53·6	5·6	9·57	3·09
	4·0	80·0	7·6	10·52	3·24
	4·5	113·9	10·2	11·16	3·34
	5·0	156·3	13·4	11·67	3·42
	5·5	208·0	17·3	12·03	3·47
	6·0	270·0	22·0	12·27	3·50
1·5	2·5	23·4	3·03	7·73	2·78
	3·0	40·5	4·14	9·78	3·13
	3·5	64·3	5·7	11·30	3·36
	4·0	96·0	7·8	12·31	3·51
	4·5	136·7	10·4	13·14	3·62
	5·0	187·5	13·8	13·58	3·68
1·75	2·5	27·3	3·08	8·87	2·98
	3·0	47·3	4·22	11·22	3·35
	3·5	75·0	5·8	12·93	3·60
	4·0	112·0	8·0	14·0	3·74
	4·5	159·5	10·7	14·9	3·86
2·0	2·5	31·3	3·13	10·0	3·16
	3·0	54·0	4·3	12·6	3·54
	3·5	85·8	6·0	14·3	3·78
	4·0	128·0	8·2	15·6	3·95
2·5	2·5	39·1	3·22	12·15	3·49
	3·0	67·5	4·5	15·1	3·89
	3·5	107·2	6·2	17·3	4·17

Considerable attention has recently been given to reducing the mass of the reciprocating parts, not only in order to lessen vibration, but also to enable higher piston speeds to be normally used, and thus diminish the weight of the engine.

At the very high piston speeds now frequent in car engines, the inertia of the piston and connecting-rod exercises an important influence upon the distribution of the driving effort at the crank-pin during the working stroke, and also upon the intensity of the crankshaft pressure in the main bearings.

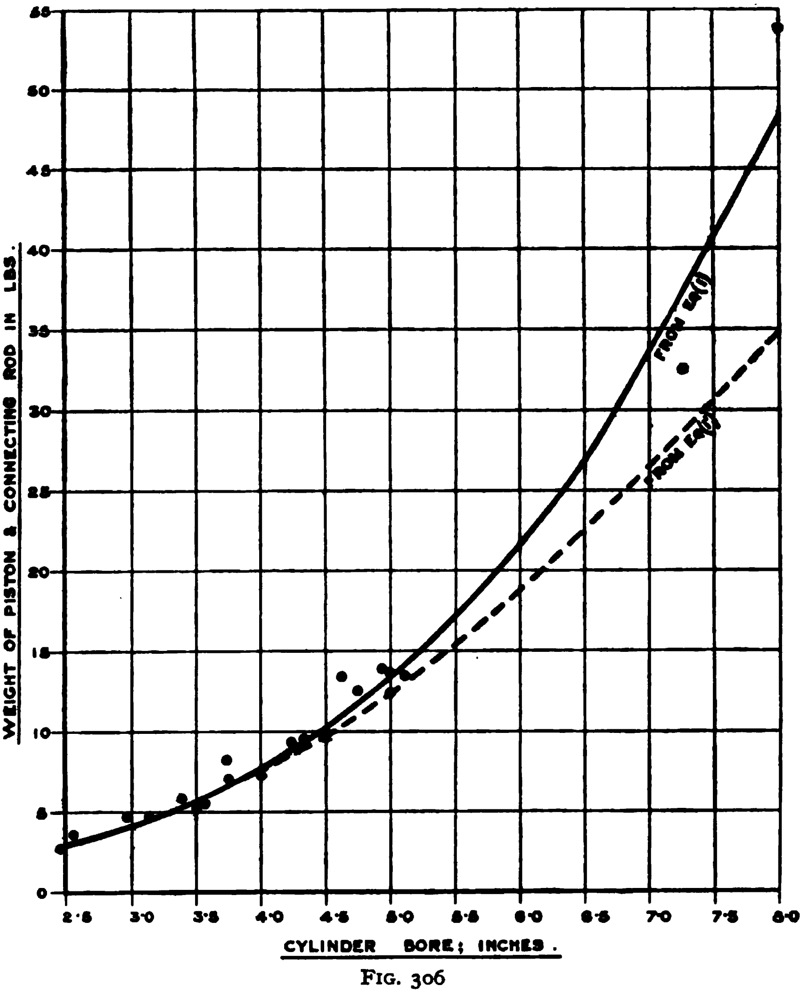


FIG. 306

Several graphical constructions exist for determining the acceleration of the reciprocating mass, among the best known being those of Klein, Rittershaus, and Mohr.

Prof. Klein's method was originally given in the *Journal of the Franklin Institute* for September and October 1891, for the general case of a quadric crank chain ; in the particular case of the crank and connecting-rod it reduces to the following simple procedure :[1]

In fig. 307 let O P, P M be any position of the combination. From H, the middle point of P M, as centre, describe an arc through P. Produce M P to meet the perpendicular to the engine axis at K. From P, as centre, with radius P K, describe an arc cutting the former at L and N ; join L N and produce, if necessary, to meet the engine axis at R.

Then O R measures the acceleration of M to the same scale in which O P measures the arcual velocity of P.

At M erect an ordinate, M Q, equal to O R ; then Q is a point on the acceleration curve ; by repeating this process other points may be obtained, and the curve of acceleration D Q C E can be traced through them.

It is usual to consider the crank-pin as moving with uniform velocity ; if *n* denote the number of revolutions per *minute*, and ω the angular velocity in radians per *second* of the crank-pin, then

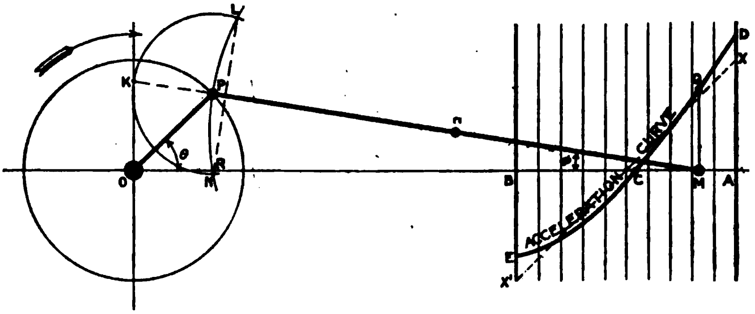


FIG. 307

$$\omega = \frac{2\pi n}{60} = \frac{\pi n}{30} = 0{\cdot}1047\,n \qquad (3)$$

It is useful to know the exact values of the acceleration when the piston is at the top and bottom of its stroke ; these are :

$$\text{At top : AD} = \omega^2\rho\left(1 + \frac{\rho}{l}\right) \text{ f.s.s.} \qquad (4)$$

$$\text{At bottom : BE} = \omega^2\rho\left(1 - \frac{\rho}{l}\right) \text{ f.s.s.} \qquad (5)$$

If the connecting-rod were indefinitely long, the point M would have a simple harmonic motion, in which case the curve of acceleration would be the straight line $x\,x^1$; the effect of the obliquity of the rod is shown by the degree in which the curve DQCE departs from this line. The acceleration tends always to the point C, at which the reciprocating parts have a maximum linear velocity along the line of stroke.

---

[1] For demonstration of the method see Appendix I.

The force necessary to produce this acceleration is obtained by multiplication by the measure of the reciprocating mass. Here it should be pointed out that as the connecting-rod partly reciprocates and partly rotates, only part of its mass should be reckoned as reciprocating, the remainder being regarded as revolving with the crank-pin; Herr Güldner considers that from 0·45 to 0·55 of the connecting-rod mass should be regarded as reciprocating; for car engines it is probably sufficiently accurate to take the fraction, roundly, at the value 0·5.

Denoting by μ the mass in lbs. of the part regarded as reciprocating, we may in usual cases, then, take $\mu = 0{\cdot}75\ m$ (see Eq. (1); *ante*).

Thus if F denote the force in lbs.-weight necessary to produce the accelerations at the top and bottom of the stroke, we have:

$$\text{At top: } \quad F_a = \frac{\mu\omega^2\rho}{g}\left(1 + \frac{\rho}{l}\right) \text{ lbs.} \qquad (6)$$

$$\text{At bottom: } \quad F_b = \frac{\mu\omega^2\rho}{g}\left(1 - \frac{\rho}{l}\right) \text{ lbs.} \qquad (7)$$

Now d denoting the piston diameter in inches, if p^1 be the pressure in lbs. per sq. in. upon it necessary to produce the gross force F, then $F = \frac{\pi}{4} d^2 p^1$, so that $p^1 = \frac{4F}{\pi d^2}$; thus from (6) we have:

$$\text{At top: } \quad p^1 = \frac{4\mu\omega^2\rho}{\pi g d^2}\left(1 + \frac{\rho}{l}\right) \text{ lbs. sq. in.} \qquad (8)$$

For practical calculations this result may be simplified; for if s be the stroke in inches, then $\rho = \frac{s}{24}$. Also, as already pointed out, $\omega = \frac{\pi n}{30}$; $\mu = 0{\cdot}75m$ (*q.p.*); while for car engines $\frac{\rho}{l}$ departs but little from the value $\frac{2}{9}$. Thus, on substitution and reduction, Eq. (8) becomes:

$$p^1 = 0{\cdot}0000166\ \frac{mn^2s}{d^2} \text{ lbs. sq. in.} \qquad (8\text{A})$$

in which d and s are in inches, n in revs. per min., and m in lbs.

Observe that as p^1 is obtained from the acceleration by multiplying by a constant expression, an alteration of scale alone enables the acceleration curve to also represent the variation in p^1 throughout the stroke. Also that the inertia of the parts involves no loss of work, the energy absorbed from the crank-pin during the portion A C of the down stroke being restored to it during the remaining portion C B; the work areas A C D and B C E are equal and opposite. It is of interest to note the large values attained by p^1 in many cases, notwithstanding the small mass of the reciprocating parts, owing to the high speed of rotation; in the following short table some results from actual engines are given:

Ref. No.	Bore in ins. *d*	Stroke in ins. *s*	Revs. per min. *n*.	Piston speed, f.p.m.	Mass, *m*, lbs.	Value of p^1 lbs. sq. in.
1	4·0	7·0	1560	1820	5·5	97
2	2·44	4·33	2000	1443	3·2	154
3	4·75	5·0	2000	1670	12·22	179
4	3·53	4·75	2400	1900	5·38	195
5	3·74	5·32	2300	2040	7·01	234
6	3·15	5·12	2900	2475	3·7	266

Referring now to fig. 308, suppose the engine No. 6 to be running and the piston at A commencing its downward suction stroke. The pressure, in lbs. per sq. in. of piston area, required to produce the necessary acceleration of the reciprocating parts is measured by A D, in this case (*v.* table) about 266 lbs. per sq. in. As, however, there is no pressure on the piston during this stroke, the necessary force must be supplied from the crank-pin; hence from A to C the crank-pin must drive the reciprocating mass. At C this mass has its maximum velocity, and is thereafter subject to a retardation of magnitude increasing from zero at C to B E at B; thus from C to B the reciprocating mass drives the crank-pin, restoring during this period the work given to it during the first period A C.

Hence at C there is a reversal of the pressure between the crank-pin and connecting-rod end.

During the return (compression) stroke, similar reasoning shows that there must be a reversal of pressure at U, where the compression and acceleration lines intersect, and that from U to A the crank-pin retards the reciprocating mass.

At A ignition occurs, and a pressure represented by A W is suddenly created on the piston; a reversal of pressure accordingly takes place immediately, the reciprocating mass driving the crank-pin, which condition continues, in normal cases, throughout the working stroke of the piston, the pressure available for producing driving effort at the crank-pin being represented at any point of the stroke by the vertical height, at that point, of the shaded diagram DCEBYW. During the exhaust stroke there is again a reversal of pressure at C.

Hence in a complete normal cycle we have:

During suction stroke, a reversal at C.
During compression, a reversal at U.
At ignition, a reversal at A.
During expansion, no reversal.
During exhaust, a reversal at C.

The highest speed of which the authors have good evidence is that of engine No. 6, which has on several occasions maintained about 2900 r.p.m. in a car for twenty consecutive minutes.

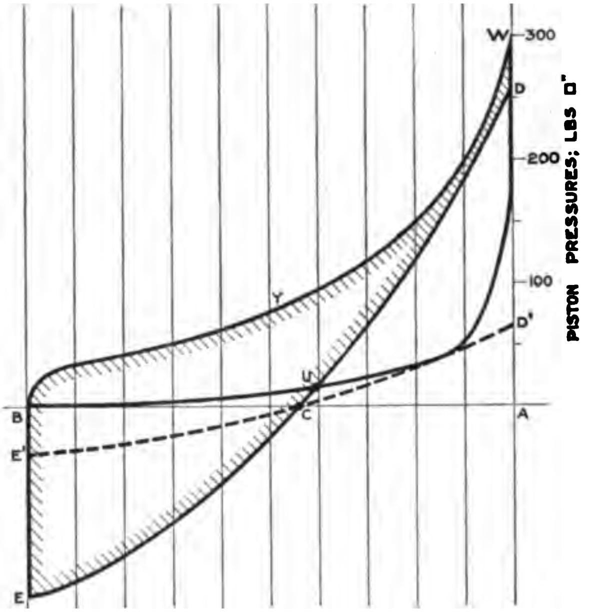


FIG. 308

In fig. 308 the case for this engine is plotted to scale, using an actual diagram (taken at a somewhat lower speed) obtained by a mirror-diaphragm indicator, and it will be noted that at the commencement of the downward working stroke, with the explosion pressure assumed, nearly all the pressure on the piston is absorbed in accelerating the reciprocating mass, only the small amount represented by DW being available for causing driving effort at the crank-pin. The resultant pressure increases during the working stroke in proportion to the depth of the shaded area, as already pointed out; thus the effect of inertia is to transfer the maximum pressure available for causing driving effort at the crank-pin to the later portion of the working stroke.

It should be noted that the height of the acceleration pressure curve, DCE, increases in proportion to the *square* of the number of revs. per min. (Eq. (8A)); thus, at 1500 revs. it sinks to the position indicated by the dotted line D^{1}CE^1 and the inertia then tends to equalise the resultant driving pressure during the working stroke.

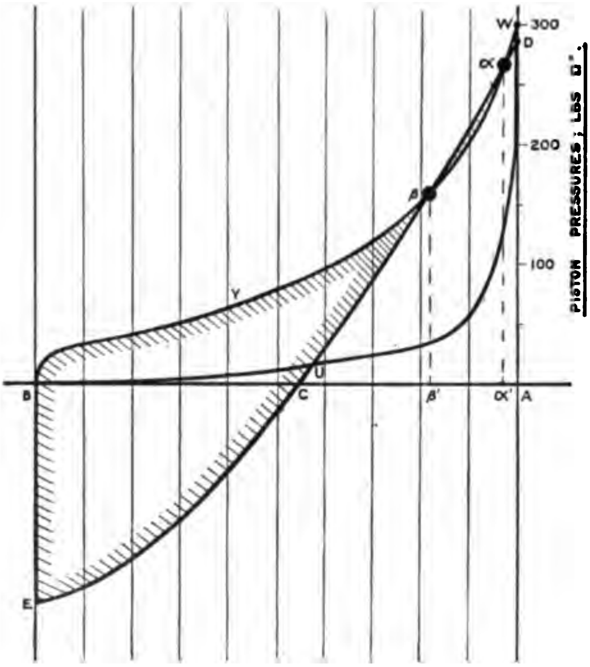


FIG. 309

On the other hand, if the engine speed were increased to about 3000 revs. per min., the acceleration pressure curve would cut the expansion line of the indicator diagram at the points α and β, as shown in fig. 309. Such cases are very rare; the authors have not, so far, met with one instance in which all the data have been so reliable as to enable one to declare with certainty that the engine has run quite so fast as to realise this condition; the condition is, however, undoubtedly very closely approached in many racing engines.

When the two curves cut there are three reversals of pressure at the crank-pin during the working stroke, viz. at D, α, and β. DαW, and βCEBY being positive work-areas, while that between a and β is negative.

This triple reversal would tend to set up additional vibration, and this consideration, coupled with the fact that this critical speed is occasionally approached but apparently very rarely exceeded by certain racing engines, has led Burls to propose that, for practical maximum power-rating purposes, the highest speed of an engine may be conveniently assumed as this critical speed.

If this be done, he points out that a formula can be deduced for the maximum BHP per cylinder, as is shown later (Eq. 21).

The maximum explosion pressure attained in petrol car engines appears to range from about 290 to 320 lbs. per sq. in. above atmosphere, and this can be realised, in good cases, at speeds up to 2000 r.p.m.

It is, however, doubtful whether at the high 'critical speed' so great an initial pressure is obtained, owing to the effects of wire-drawing and imperfect scavenging. Accordingly, Burls takes 275 lbs. per sq. in. as the maximum explosion pressure practically realisable at the limit, and that p^1 (*v.* Eq. (8A)) is about 0·95 of this, when the acceleration pressure curve will cut the expansion line as in fig. 309. Hence the maximum value of p^1 for practical cases is $275 \times 0{\cdot}95 =$ about 260 lbs. per sq. in.

Substituting, then, the value 260 for p^1 in Eq. (8A) we obtain on reduction as the maximum revolution speed:

$$n_{\text{max.}} = 3957 \sqrt{\frac{d^3}{ms}} \text{ r.p.m.} \qquad (9)$$

And if σ denote piston speed in feet per min., then

$$\sigma = \frac{ns}{6} \qquad (10)$$

so that, putting for n the value in Eq. (9), we have as the expression for maximum piston speed on Burls' assumption:

$$\sigma_{\text{max.}} = 660 \sqrt{\frac{d^3 s}{m}} \text{ f.p.m.} \qquad (11)$$

In the following table values of $n_{\text{max.}}$ and $\sigma_{\text{max.}}$, calculated from Eqs. (9) and (11), are exhibited for the range of sizes at present employed in petrol engines for car work, and it will be noted that the values obtained are not unreasonably high viewed in the light of recent racing performances.

The increase of maximum piston speed with r and d is also shown by the curves in fig. 310, which have been plotted from this table.

Table of Maximum Piston Speed and Revolutions for Car Engines with Cast-iron Pistons from Eqs. (9) and (11)

$r =$	d in ins.	$\sigma_{\text{max.}}$ f.p.m.	$n_{\text{max.}}$ r.p.m.
0·75	4·0	1700	3400
	4·5	1760	3140
	5·0	1800	2880
	5·5	1825	2650
	6·0	1840	2450
1·0	3·0	1710	3430
	3·5	1860	3190
	4·0	1940	2910
	4·5	2000	2680
	5·0	2040	2450
	5·5	2080	2270
	6·0	2100	2100
1·25	3·0	1890	3020
	3·5	2040	2800
	4·0	2130	2550
	4·5	2200	2360
	5·0	2260	2170
	5·5	2280	2000
	6·0	2300	1850
1·50	2·5	1830	2930
	3·0	2060	2750
	3·5	2210	2530
	4·0	2310	2310
	4·5	2380	2110
	5·0	2430	1940
1·75	2·5	1960	2700
	3·0	2200	2520
	3·5	2370	2330
	4·0	2460	2110
	4·5	2540	1940
2·0	2·5	2080	2500
	3·0	2330	2330
	3·5	2490	2140
	4·0	2600	1950
2·5	2·5	2300	2210
	3·0	2560	2050
	3·5	2740	1890
	4·0	2850	1710

It will be noted that on this assumption the piston speed increases both with the stroke-bore ratio, and also with the cylinder bore throughout.

In the larger sizes of petrol engines for cars, it is unusual in practice to find the stroke much exceeding the bore, although a higher piston speed would thus become practicable. This is due partly to the fact that with a long stroke a very heavy flywheel becomes necessary in order to preserve sufficient steadiness in running to provide the necessary comfort for the passengers ; and partly from difficulties of design in the transmission gearing arising from the greater engine torque. For stationary engines these considerations do not apply, and a larger value of the stroke-bore ratio could probably be used with advantage in such cases.

Horse-power Formulæ.—For an internal combustion single-acting engine, working on the Otto cycle, the BHP is exactly given by the formula :

$$\text{BHP per cylinder} = \frac{1}{33{,}000 \times 24} \cdot \frac{\pi}{4} d^2 \eta p n s \qquad (13)$$

where

d and s are the bore and stroke respectively in ins. ;

η is the mechanical efficiency of the engine ;

p is the m.e.p. in lbs. per sq. in. during the working stroke ; and n is the number of revs. per min.

If σ denote the piston speed, in feet per min., then (13) becomes, from Eq. (10) :

$$\text{BHP per cylinder} = \frac{1}{168,000} \cdot d^2\eta p\sigma \qquad (14)$$

Now it is a matter of very considerable difficulty and delicacy, outside a laboratory, to obtain accurate indicator diagrams from a fast-running petrol engine, and hence p cannot usually be directly determined ; η can be approximately determined without much difficulty if required, but in general the value of the product ηp, termed the ' mean pressure corresponding to the BHP ' or the ' brake mean effective pressure,' is found from bench tests of the BHP by aid of Eq. (13).

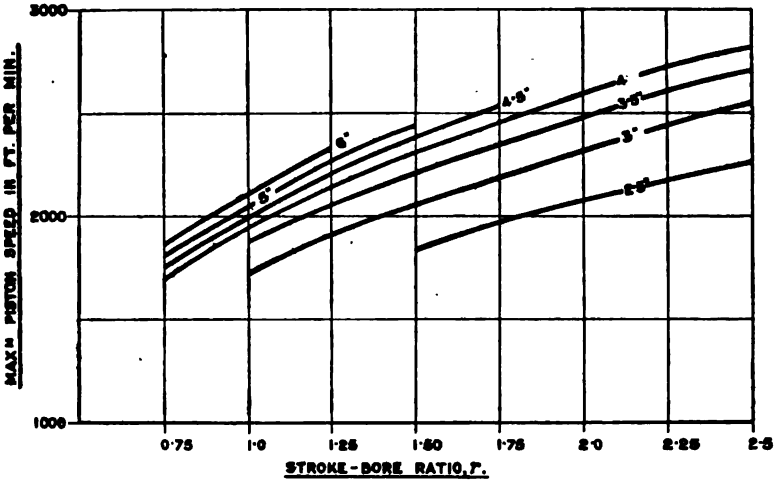


FIG. 310

For rating purposes in car and boat competitions it is important to have, if possible, a formula in terms of some readily ascertainable dimensions of the engine, which shall express, to a reasonable degree of approximation, the maximum practicable HP per cylinder.

A careful examination of nearly one hundred car engines in 1905–6 showed that the piston speed at that time might be regarded as having a constant maximum value of roundly 1000 f.p.m. ; the value of ηp was also found to average about 70 lbs. per sq. in., with little variation.

Substituting these values in Eq. (14) we have :

$$\text{BHP per cyl.} = 0{\cdot}417\, d^2 \qquad (15)$$

and accordingly the author was led to propose this expression for general rating purposes.

The R.A.C. Formula.—The Royal Automobile Club in 1906, after consideration, selected 0·4 as the value of the coefficient, and thus was obtained the well-known R.A.C. rating rule, which for long has been very widely used, and is still the basis of rating for taxation purposes, viz.:

$$\text{R.A.C. rating} = 0{\cdot}4\, d^2\text{N} \qquad (16)$$

N being the number of cylinders.

The coefficient 0·4 corresponds to a piston speed of 1000 f.p.m., and a value of 67·2 lbs. per sq. in. for ηp.

This expression is very simple in form, and for every-day rating and cataloguing purposes has proved to be very useful.

The general increase in piston speed, and the attainment of higher mean pressures in recent years due to general improvements in design and in the materials of construction, rendered it desirable that a new rating formula for competitive events should be devised, involving the stroke in addition to the bore. Many such formulæ have been proposed, a few of which may be mentioned.

Thus, a formula in terms of d and s has already been obtained, on a certain assumption, for the maximum practicable piston speed (Eq. (11)), and if the variation of ηp can be similarly expressed, it is apparent from Eq. (14) that the maximum BHP per cylinder will also be expressed in terms of d and s. For combustion chambers of similar form, the surface to which the hot gases are exposed increases as d^2, while the cubic contents of the chamber increase as d^3; hence the proportion of surface to volume is as 1 to d, and accordingly diminishes as the bore increases. Owing to the smaller heat loss, therefore, higher mean pressures may reasonably be expected, other things being equal, from a large than from a small cylinder.

But in practice richer mixtures and until recently higher compressions[1] were generally used in smaller engines, and this tended to compensate for the reduction in pressure that would otherwise probably have occurred.

In Vol. I, Chap. IV, the author has briefly considered the question of heat exchange, and has pointed out that, so far as heat loss is concerned, the Otto cycle engine with complete expansion is the best type, as exposing a given volume of working gas to the smallest cooling surface in its performance.

The whole question from a theoretic standpoint is still obscure, due to lack of experimental data. The author has maintained (*ibid.*) that it is very difficult even to say whether it is better to work with a hot or a cool cylinder. If the cylinder be kept cool, the gases lose heat more rapidly, whereas if kept hot the efficiency of the engine is reduced. In most types of gas engine, including car petrol engines, it is found practically most economical to keep the working cylinders as hot as possible; in car engines it is not unusual in winter to dismantle the radiator fan, or blank off some of the radiator tubes to reduce the cooling effect. On the other hand, in very large gas engines the water jackets are kept as cool as possible in order to limit the stresses due to unequal expansion of the metal walls. In such engines temperature rise in water jackets is dangerous to the structure.

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[1] See the table on p. 544 for recent practice.

In car engines the value of ηp usually falls off very markedly at high speeds, due to incomplete charge from wire-drawing, &c. This reduction is, however, avoidable by using sufficiently large valves and piping. Thus Mr. Remington has pointed out that by adopting suitable proportions of carburettor, induction and exhaust piping and valves, it is possible to fill and scavenge an engine running at very high piston speeds so as to obtain as high or higher a mean effective pressure as is usually associated with moderate or even low piston speeds.

No direct experiments have been made with a view to ascertain the nature of the dependence of ηp upon d and s; any such experiments would be lengthy and costly, as so many considerations are involved. The indirect evidence, however, is sufficient to render it very probable that, on the whole, an increase of ηp with d should occur in comparable cases.

Thus, Prof. Callendar (*Proc. Inst. A.E.*, 1906–7), from the results of some experiments over a limited range, was led to propose the relation

$$\eta p = k\left(1 - \frac{1}{d}\right) \tag{17}$$

as well representing the observed facts. The rating Sub-Committee of the Society of Motor Manufacturers and Traders, after consideration of a number of further cases, adopted an expression of this form in their Report on HP formulæ issued in August 1908.

In a subsequent discussion of this Report by the Institution of Automobile Engineers, Prof. Callendar said:

'The experiments of the Automobile Club of France on 96 engines ranging very equally from 65 mm. to 190 mm. (2½ ins. to 7½ ins.) in bore . . . are practically conclusive evidence of the increase of mean pressure with size.'

And further (*ibid.*):

'We cannot be far wrong if we take the formula $cd\,(d - 1)$, which allows only half the rate of increase observed by the A.C.F., . . . to represent the increase of ηp with d.'

Mr. Remington (*ibid.*) says:

'From an examination of the normal tests of the Wolseley engines many lessons have been learned, conspicuous among them being the increase of brake mean effective pressure with increase of size.'

Lastly, Mr. Poppe (*ibid.*):

'We have now tested over 1700 engines of bores from 3·15 ins. to 5 ins.; the formula we have arrived at for the calculation of BHP at a piston speed of about 1000 f.p.m. is:

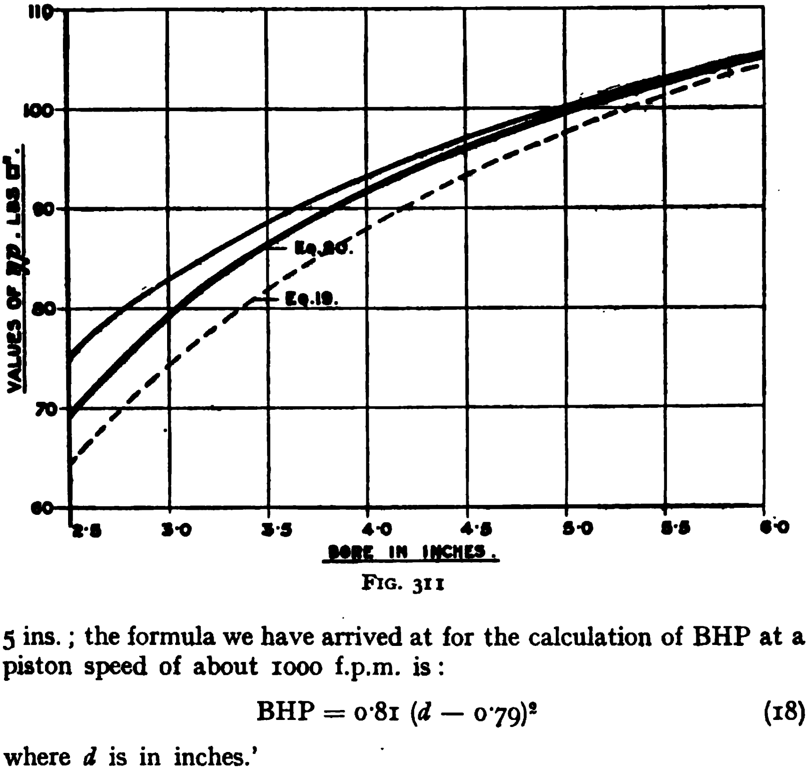


Fig. 311

$$\text{BHP} = 0{\cdot}81\,(d - 0{\cdot}79)^2 \qquad (18)$$

where *d* is in inches.'

Comparing this with Eq. (14), we see its form implies that

$$\eta p = 136\left(1 - \frac{0{\cdot}79}{d}\right)^2 \qquad (19)$$

and hence that in these trials ηp has increased with increase of cylinder bore; the following are results from this formula:

$d =$	$2\frac{1}{2}''$	$3''$	$4''$	$5''$	$6''$
$\eta p =$	63·5	74	87	96	102

The following table shows the results obtained from tests of a series of engines built by the same firm; it will be noted that the value of ηp steadily rises as the engines increase in size:

d, inches	s, inches	n, r.p.m.	BHP by test	ηp, lbs. per sq. in.
3·15	3·54	1670	18·7	80·6
3·35	4·33	1360	22·6	86·2
3·94	4·33	1360	32·0	88·2
3·94	5·12	1155	33·3	91·8
4·33	5·12	1155	40·8	93·0
4·73	5·12	1155	50·6	96·5
5·0	5·12	1155	59·5	102·0

The piston speed is in each case about 1000 f.p.m.; plotting the values of ηp against d, we get the curves shown by the upper full line in fig. 311; the dotted line shows the variation in accordance with Eq. (19) representing Mr. Poppe's experience.

The particular series of engines selected thus give values rather higher, especially for smaller engines, than those furnished by Eq. (19).

Burls (*Proc. Inst. A.E.*, Vol. V), from an examination of test results from about sixty engines, found that ηp appeared to increase with size; the relation best resuming the figures obtained was:

$$\eta p = 130\left(1 - \frac{1}{0{\cdot}85d}\right) \qquad (20)$$

From this the following values result:

d	=	2½″	3″	4″	5″	6″
ηp	=	69	79	91½	99½	104½ lbs. per sq. in.

and on comparing it will be seen that they are in practical agreement with the figures obtained from Eq. (19); in fig. 311 these values are shown by the middle curve. The relation between ηp and d expressed in Eq. (20) was adopted in the rating formula recommended by the Joint Committee of the Royal Automobile Club, the Inst. A.E., and the Soc. of Motor Manufacturers in their Report issued in Feb. 1911. As it is simpler in form than Eq. (19) and correct to an equal degree of approximation, it is here used as expressing the possible variation of ηp with bore in engines designed with equal skill.

Burls' Max. HP Formula.—Taking, then, Eqs. (11) and (20), and substituting for σ and ηp respectively in Eq. (14), we obtain on reduction Burls' suggested rating formula in d, s, and m for maximum BHP per cylinder, viz.:

$$\text{Max. BHP per cyl.} = \tfrac{1}{2}d\,(d - 1{\cdot}18)\sqrt{\frac{d^2 s}{m}} \qquad (21)$$

The following table calculated from this equation gives figures for the maximum BHP per cylinder for the range of sizes usual in car engine practice, the values of $\frac{d^2 s}{m}$ being taken from the table on p. 507 (*ante*); in fig. 312 curves are also shown, plotted from the figures of the table.

MAXIMUM BHP PER CYLINDER. FROM EQ. (21). CAST-IRON PISTONS

$r =$	$d =$	Max. BHP	$r =$	$d =$	Max. BHP
0·75	4·0	14·5	1·50	2·5	4·6
	4·5	20·0		3·0	8·6
	5·0	26·0		3·5	13·6
	5·5	33·0		4·0	19·8
	6·0	40·5		4·5	27·0
1·0	3·0	7·0		5·0	35·2
	3·5	11·5	1·75	2·5	4·9
	4·0	16·5		3·0	9·2
	4·5	22·7		3·5	14·7
	5·0	29·5		4·0	21·2
	5·5	37·5		4·5	28·8
	6·0	46·0	2·0	2·5	5·2
1·25	3·0	7·9		3·0	9·6
	3·5	12·5		3·5	15·4
	4·0	18·3		4·0	22·2
	4·5	24·9	2·5	2·5	5·7
	5·0	32·7		3·0	10·6
	5·5	41·2		3·5	17·0
	6·0	50·6		4·0	24·4

The results from Eq. (21) are very high, and are only approximated to at present for short periods of running and under the special conditions of racing where every refinement of practice is utilised. They appear, however, to be not unreasonably high when compared with some actual engine performances.

Thus a four-cylinder 16–20 engine by the Sunbeam Co. developed on test 13·56 BHP per cylinder at 2300 r.p.m.

Here $d = 3{\cdot}74$ ins., $s = 5{\cdot}32$ ins., and $m = 7{\cdot}1$ lbs.; whence Eq. (21) gives 15·5 as the maximum value; the actual power was thus 87 per cent. of the estimated maximum in this case.

Again, referring to the 100 mm. × 250 mm. (3·94 ins. × 9·85 ins.) single-cylinder de Dion racing engine, the *Autocar* for May 14, 1910, said:

'It is claimed that this engine will develop about 35 HP, but it is probable that 30 will be nearer the mark.'

Here:

$d =$	$s =$	$r =$	$m =$	$\sqrt{\frac{d^2s}{m}} =$
3·94	9·85	2·5	5·7	5·18

whence

$$\sigma_{\text{max.}} = 3420 \text{ f.p.m.} \qquad n_{\text{max.}} = 2080 \text{ r.p.m.}$$

while from Eq. (21):

$$\text{Max. BHP} = 28$$

a result in satisfactory agreement with the anticipated performance.

This de Dion engine was fitted to the Le Gui and Werner cars in the Coupe des Voiturettes competition of June 1909; the competition was won by the Le Gui, which covered the whole course of roundly 280 miles at an average speed of 47½ miles per hour.

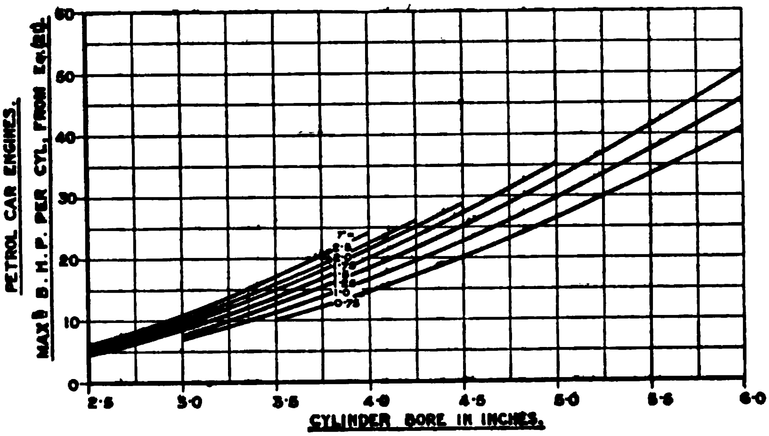


FIG. 312

Commenting on this, the *Automotor Journal* remarked that it was 'another case of victory for the single-cylindered, long-stroke racing engine.'

These engines had hemispherical combustion chambers with the exhaust proceeding from the cylinder top. An unusually high compression ratio was used.

The 1908 Coupe des Voiturettes competition was won by a Delage car fitted with a single-cylindered engine of 3·94 ins. bore, and 5·92 ins. stroke. Here $\sqrt{\frac{d^2s}{m}} = 4{\cdot}18$, and hence, by Eq. (21):

$$\text{Max. BHP} = 22{\cdot}7$$

A rough check upon this figure may be obtained as follows:

The area, A, opposed by the car to wind resistance was about 12 sq. ft., and the gross weight, W, about $\frac{2}{3}$ ton. The road resistance, R, may be taken at the usual summer figure of 50 lbs. per ton; and the car covered the whole course at an average speed of just 50 miles per hour. Hence, using Stanton's result (*Proc. Inst. C.E.*, Vol. CLXXXI, p. 191) we have, V denoting speed in miles per hour:

$$\begin{aligned}\text{Eff. HP at wheels} &= \mathrm{V}\left(\frac{\mathrm{WR}}{375} + \frac{8{\cdot}54}{10^6}\mathrm{AV}^2\right)\\ &= 4{\cdot}4 + 12{\cdot}8\\ &= 17{\cdot}2 \text{ roundly}\end{aligned}$$

At top speed there is one gear transmission only between the flywheel and the road wheels; taking the efficiency of this as 0·8, we have:

$$\text{Average BHP} = \frac{17{\cdot}2}{0{\cdot}8} = 21{\cdot}5$$

which supports the figure obtained from Eq. (21).

Again, as an instance of a large engine with an unusually high value of r, consider the 1910 type four-cylinder, 130 mm. × 190 mm., 90 HP F.I.A.T. engine.

Here

$$d = 5{\cdot}12 \qquad s = 7{\cdot}48$$

Hence, by Eq. (1), for cast-iron pistons:

$$m = 14{\cdot}6 \text{ lbs., and } \sqrt{\frac{d^2s}{m}} = 3{\cdot}66$$

Substituting in Eq. (21) we have as the maximum BHP per cylinder for this case:

$$\text{Max. BHP} = 36{\cdot}8$$

a result practically identical with the stated maximum output of $\frac{143}{4} = 35{\cdot}8$ BHP per cylinder.

In the *Autocar* for October 29, 1910, there appeared an illustration and note on the performance of a four-cylinder White & Poppe engine, 80 mm. × 130 mm. (3·15 ins. × 5·12 ins.), having an R.A.C. rating of 15·9. An 'all out' test of this engine furnished the following results:

R.P.M. =	1900	2000	2200
BHP =	38	44	47

Thus the maximum HP per cylinder was 11·75.

The actual mass, m, of the reciprocating parts is not stated; if this be estimated by aid of Eqs. (1) and (2) of pp. 504 and 506 we get:

For steel pistons: $m = 3{\cdot}44$ lbs.

For cast-iron pistons: $m = 4{\cdot}6$ lbs.

Hence, $\sqrt{\frac{d^2s}{m}}$ has in this case the value 3·84 for steel, and 3·32 for cast-iron, pistons.

Using Eq. (21) we have, therefore, for the maximum power rating:

$$\text{Max. BHP per cylinder} = \begin{cases} 11{\cdot}9 \text{ with steel pistons,} \\ 10{\cdot}3 \text{ with cast-iron pistons,} \end{cases}$$

a result again in satisfactory accord with the actual performance.

The four-cylinder 3·54 ins. × 4·73 ins. Vauxhall engine in 1910 developed 13 BHP per cylinder at 2400 r.p.m. with cast-iron pistons.

Writing in January, 1911, Mr. L. H. Pomeroy said:

'We have now raised this to 15 BHP per cylinder at 2500 revs. per min., mainly by reduction in reciprocating weight and improvement in mechanical efficiency.'

In these later tests the pistons were of steel. Applying the formula (21) to this case we have, using firstly the equation for cast-iron pistons (1):

$$\text{Max. BHP per cyl.} = 13{\cdot}5$$

while with m for steel pistons, Eq. (2):

$$\text{Max. BHP per cyl.} = 15{\cdot}8$$

again in both cases satisfactory ratings.

Thus it is clear that under racing conditions the high power ratings given by the semi-empirical Eq. (21) are already closely approached in certain cases.

To obtain from this equation the rating of any given engine, it is, strictly speaking, necessary to weigh the reciprocating parts in order to determine m. This is not, however, absolutely necessary, as m can be found with sufficient accuracy from d and s by aid of Eqs. (1) and (2). A table could readily be constructed exhibiting values of $\sqrt{\frac{d^2s}{m}}$ for engines with cast-iron, and with pressed steel, pistons and connecting-rods, for use in this connection.

The S.M.M.T. Formula.—The Society of Motor Manufacturers and Traders, in a Report on horse-power formulæ, dated August 1908, recommended for touring car engines the Callendar type formula:

$$\text{BHP per cyl.} = 0{\cdot}197\, d\,(d - 1)\,(r + 2) \qquad (22)$$

the coefficient being increased to 0·333 for racing engines.

This involves the assumptions: (A) that the maximum practicable piston speed slowly increases with the stroke-bore ratio r, agreeably with the relation:

$$\sigma = \frac{1000}{3}\,(r + 2) \text{ f.p.m.}$$

and (B) that the brake mean effective pressure, ηp, increases with the bore according to the Callendar formula: $\eta p = 99 \left(1 - \frac{1}{d}\right)$ lbs. per sq. in. In both cases a slow rate of increase was taken, the rise in value of ηp with d, for example, being only about one-half as great as that observed in the A.C.F. engines already referred to. Eq. (22) is a useful rating formula for the purpose recommended.

The Midland A.C. Formula.—The formula of the Midland Automobile Club proved useful, and has been much used in the past for rating in car competitions ; this formula is :

$$\text{Power rating per cyl.} = \text{K}d^2\sqrt[3]{s} \tag{23}$$

where K is a constant.

The good results given by this expression are due to a curious double coincidence, as Prof. Callendar has pointed out ; for the expression may be written :

$$\text{K}d^2\sqrt[3]{d}\sqrt[3]{r}$$

Comparing with Eq. (14), this implies that ηp varies as $\sqrt[3]{d}$, and that σ varies as $\sqrt[3]{r}$, both implications being practically in accord with fact prior to the recent marked increase in pressures and piston speeds.

Prof. Henderson proposed a formula in which the fundamental linear quantity is the cube root of the piston displacement volume, viz. $\sqrt[3]{d^2s}$.

His formula may accordingly be written :

$$\text{Power rating per cyl.} = \text{K}d^2r^{\frac{2}{3}} \tag{24}$$

Mr. F. W. Lanchester (*Proc. Inst. A.E.*, 1906–7) concludes that the maximum practicable piston speed increases with the stroke-bore ratio—a conclusion also reached independently by Prof. Callendar—and that a formula of the general type

$$\text{BHP per cyl.} = \text{K}d^ns^{2-n} \tag{25}$$

is competent for rating purposes.

He suggested, at that date, that the value of n best suited to current practice was 1·5, with a value of 0·4 for K ; hence his complete expression was :

$$\text{BHP per cyl.} = 0{\cdot}4\, d^2\sqrt{r} \tag{26}$$

Prof. Henderson later modified this equation with a view to facilitating computation, and expressed it in the simpler approximate form :

$$\text{BHP per cyl.} = 0{\cdot}2\, d^2(r + 1) \tag{27}$$

Displacement Formula.—A formula that was used a good deal for general purposes of car engine classification in past years is based upon the total volume swept through by the piston per min.; this quantity is $\kappa d^2 sn$, a usual value taken for κ being $\frac{1}{12,720}$, thus:

$$\text{`Displacement' rating} = \frac{1}{12,720} \cdot d^2 sn \qquad (28)$$

Comparison with Eq. (13) shows that this implies ηp as of the constant value 79·25 lbs. per sq. in. The displacement rating involves a knowledge of n, if absolute results are required.

An expression that has long been used in drawing offices for roughly checking designs is:

$$\text{BHP per cyl.} = \kappa d^2 s \qquad (29)$$

and a formula of this type has recently been supported by Mr. Dendy Marshall and others for use in rating, the value of κ being taken as $\frac{1}{12}$ (*v.* the *Autocar*, June 5, 1909).

Comparison with the exact Eq. (13) shows this to involve the assumption that

$$\eta p = \frac{84,000}{n}$$

which is independent of the bore. This assumption is largely gratuitous.

The M.M.A. Formula.—The Marine Motor Association adopted for the petrol engines of boats in 1910 a rule as follows:

$$\text{Rating per cyl.} = \text{area of exhaust valve} \times 3\tfrac{1}{4} \qquad (30)$$

As will be seen shortly, this rule furnishes ratings of the same type as those given by the R.A.C. rule when the valves are regularly proportioned with reference to the bore and stroke of the engine.

The Joint Committee Formula.—A joint committee on the horse-power formula problem, composed of representatives of the R.A.C., the Inst.A.E. and the S.M.M.T., sat under the chairmanship of the author during 1909–1910, and a Report was issued in February 1911 by this body, in which the following is recommended as a rating formula for maximum bench-test performance per cylinder:

$$\text{Rating} = 0{\cdot}45\,(d + s)\,(d - 1{\cdot}18) \qquad (22\text{A})$$

This is of the same form in the variables as the Callendar type S.M.M.T. formula (22); it involves an increase in maximum practicable piston speed with stroke-bore ratio as expressed by the relation $\sigma = 600\,(r + 1)$ f.p.m., and an increase of ηp with bore in accordance with Eq. (20), as already mentioned. The ratings furnished by this formula are conveniently shown in the accompanying diagram, fig. 313.

In an able paper read before the Inst. of Automobile Engineers in December 1911, Mr. L. H. Pomeroy, of the Vauxhall Motor Co., supported the displacement formula for rating purposes in competitions.

He points out that large power output depends upon high mean effective pressure, high speed, and high mechanical efficiency, and that these in their turn result from:

(1) The use of richly carburetted mixtures giving considerable volume increase in the products of combustion (*v.* Chap. IX, fig. 354).

(2) High compression, ensuring prompt ignition at high revolution speeds.

(3) High volumetric efficiency obtained by the use of adequately large valves and ports; the petrol engine should be carefully designed as an efficient *pump* as well as from the point of view of a prime mover.

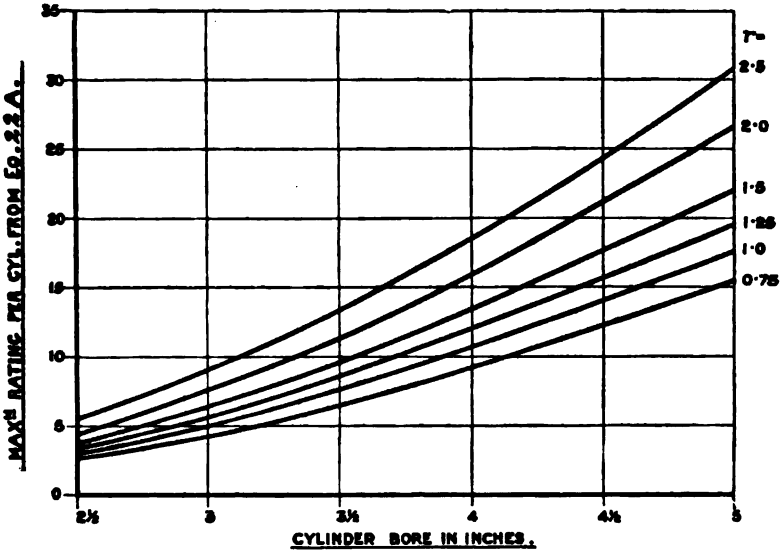


FIG. 313

(4) Small mass of reciprocating parts conducing to high speed and high mechanical efficiency.

The power output will then, *cæteris paribus*, be proportional to the revolution speed, and that engine which, by skilful design, is able to run at the highest revolution speed will then develop the greatest power. The displacement formula has commended itself also to the Brooklands Automobile Racing Club; this body, in February 1912, adopted a table of standard classes for the petrol engines of motor cars based upon the volume, $\frac{\pi}{4}d^2 s \text{N}$, swept through by the engine pistons; N denotes the number of cylinders. The following table issued by the club gives limiting bore and stroke dimensions for four-cylinder, single-acting Otto cycle engines in accordance with this formula:

BROOKLANDS STANDARD ENGINE CLASSES, BASED ON 'CYLINDER CAPACITY' $\frac{\pi}{4}d^2sN$ (*February* 1912)

Letter of class	A	B	C	D	E	F	G	H	J
*Maximum cylinder capacity in cubic inches	100	125	150	175	235	305	475	850	over 850
*Maximum cylinder capacity in cubic centimetres	1639	2048	2458	2868	3851	4998	7784	13929	over 13929
*Minimum cylinder capacity in cubic inches	—	101	126	151	176	236	306	476	851
*Minimum cylinder capacity in cubic centimetres	—	1640	2049	2459	2869	3852	4999	7785	13930
†Weight in lbs.	1400	1500	1600	1800	2000	2250	2500	2700	3000
Maximum stroke in ins. for a bore of 2½ ins. (4 cyl.)	5·118	6·391	—	—	—	—	—	—	—
,, ,, mm. ,, 65 mm. (,,)	123·5	154·3	—	—	—	—	—	—	—
,, ,, mm. ,, 70 mm. (,,)	106·5	133·0	159·7	—	—	—	—	—	—
,, ,, mm. ,, 75 mm. (,,)	92·7	115·9	139·1	162·3	—	—	—	—	—
,, ,, ins. ,, 3 ins. (,,)	3·554	4·438	5·322	6·207	—	—	—	—	—
,, ,, mm. ,, 80 mm. (,,)	81·5	101·8	122·2	142·6	191·5	—	—	—	—
,, ,, mm. ,, 85 mm. (,,)	—	90·2	108·3	128·3	169·7	—	—	—	—
,, ,, ins. ,, 3½ ins. (,,)	—	3·261	3·910	4·560	6·119	—	—	—	—
,, ,, mm. ,, 90 mm. (,,)	—	—	96·6	112·7	151·4	196·4	—	—	—
,, ,, mm. ,, 100 mm. (,,)	—	—	—	91·3	122·6	159·1	—	—	—
,, ,, ins. ,, 4 ins. (,,)	—	—	—	3·491	4·685	6·077	—	—	—
,, ,, mm. ,, 110 mm. (,,)	—	—	—	75·4	101·3	131·4	—	—	—
,, ,, ins. ,, 5 ins. (,,)	—	—	—	—	—	3·889	6·054	—	—
,, ,, ins. ,, 6 ins. (,,)	—	—	—	—	—	—	4·204	7·520	—
,, ,, mm. ,, 155 mm. (,,)	—	—	—	—	—	—	103·1	184·5	—
,, ,, ,, ,, 190 mm. (,,)	—	—	—	—	—	—	—	122·8	—

Fractions of a cubic inch or cubic centimetre will be reckoned as 0·0 if below 0·5, and as 1·0 if 0·5 or over. † Minimum weight, includes driver.

NOTE.—1 cub. in. = 16·38706 cub. cent. 1 cub. cent. = 0·0610237 cub. inch.

VALVES

Excepting only the Daimler-Knight sliding valve system, the poppet valve is at present almost universally used in petrol engines for cars.

Poppet valves are in general arranged with their axes parallel to that of the cylinder; in the Lanchester engine, however, as will be noted on reference to fig. 351, Chap. VIII, the valve axes are at right angles to that of the cylinder.

When automatic inlet valves were used the common arrangement was to place both valves in a side pocket of the combustion chamber, the exhaust being driven from below by a cam and tappet-rod from the half-speed shaft, while the inlet was placed directly above it.

This enabled valves of adequate size to be fitted without involving a wide pocket; it was convenient for machining, and for inlet- and exhaust-pipe arrangement. In several recent designs, e.g. the 20 HP Adler engine (fig. 289), a return to this early arrangement appears, excepting, of course, that the inlet valve is now mechanically operated by overhead gear; the advantages of low lift valves of large diameter are thus obtained, together with a compact form of combustion chamber and short engine. The arrangements most usually met with are, however:

(A) That in which both valves are on one side of the cylinder, arranged side by side in the same pocket; and

(B) That in which the valves are on opposite sides of the cylinder, each in its own pocket.

This latter arrangement necessitates a second half-speed shaft and actuating gear-wheel and gives a larger surface to the combustion chamber. It possesses, however, the practical advantages:

(1) Of enabling a very convenient arrangement of the exhaust and inlet piping to be made;

(2) Of giving a short engine, especially where the cylinders are cast separately; and

(3) In the opinion of many designers it gives a cleaner and better scavenged engine than (A).

With the (A) arrangement—very usual in two-cylinder engines, cast together—in order to accommodate valves of adequate diameter the valve pocket may require to be made wider than the cylinder, and hence when four or six cylinders are used the engine will necessarily be longer than with disposition (B).

In a few engines, e.g. the Maudslay, the valves are contained in the cylinder head, necessitating some type of overhead driving gear. It is not so easy to get in valves of large diameter in this position as when arranged in one or two pockets; on the other hand, the inflow and outflow of gas past the valves is here probably freer than in the more usual arrangement wherein it frequently happens that a considerable fraction of the valve opening towards the wall of the pocket is of little or no use. For this reason valves of smaller diameter placed in the cylinder head may prove practically equal to larger valves situated in pockets. It is frequently found that a less flexible engine is obtained with overhead valves, due probably, as Dr. Watson has conjectured, to the working charge in the vicinity of the firing spark being then of average richness only instead of, as in the more usual arrangement, more than average richness near this point. With overhead valves it has sometimes been found necessary to adjust the carburettor to give a richer mixture in order to obtain regular ignition at low speeds.

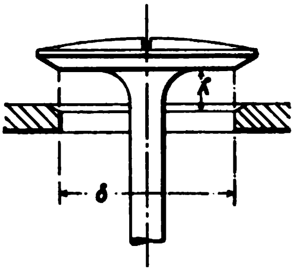


FIG. 314

In the Lanchester arrangement, with the valve axes horizontal, the valves may be of adequate diameter and the point of ignition near the inlet; from the constructional point of view, however, a vertical disposition of the valves is usually considered preferable.

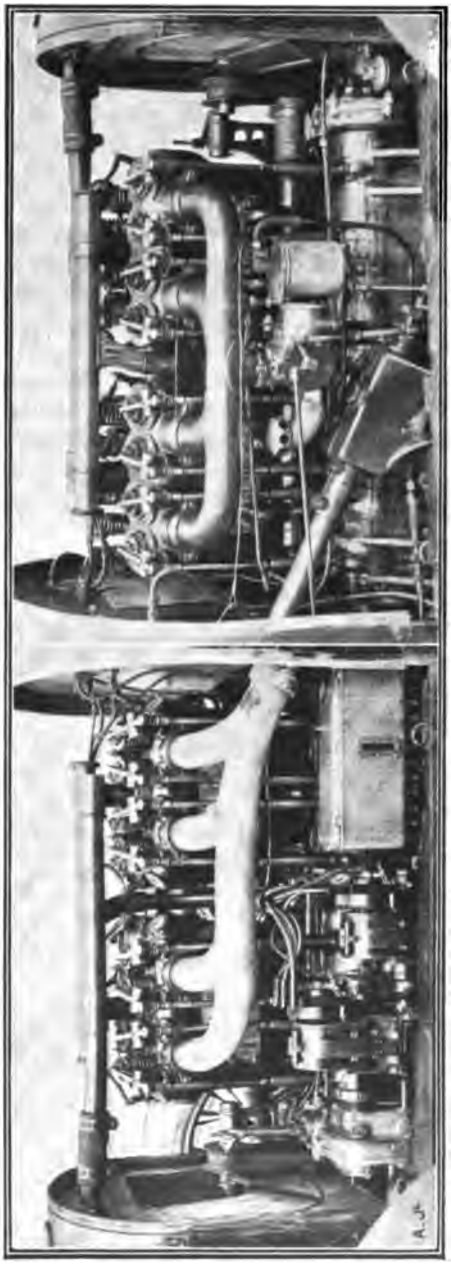
Valves are usually of mild or nickel steel, with coned seats having an angle of about 30°. Now that inlet valves are all mechanically operated, the most usual practice is to make the inlets and exhausts of the same size, so that they may be interchangeable.

But as cool gas is more viscous than hot gas, and as, further, there is usually less pressure head causing flow through the inlet valve than through the exhaust, it is probably good practice to make the inlets somewhat larger in diameter than the exhausts, as is done in some of Messrs. White & Poppe's engines.

If δ be the diameter of the valve throat, fig. 314, and λ the lift of the valve, a cylindric surface $\pi\delta\lambda$ (app.) is available for the passage of gas normally across it; if we equate this to the throat area $\frac{\pi}{4}\delta^2$ we have on reduction

$$\lambda = \tfrac{1}{4}\delta$$

as the valve lift corresponding to diameter δ. So great a lift as one-quarter of the valve diameter is unusual in car engine practice on account of inertia effects; at very high speeds the tappet rod parts contact with the cam even though very stiff springs be employed. To overcome this difficulty and obtain sufficient inlet and outlet area at very high speeds some racing engines have been fitted with two inlet and two exhaust valves to each cylinder as, for example, the Benz engine illustrated in fig. 315.



[*Yellow Cover, Copyright*].

FIG. 315.—Benz Racing Engine, with Two Inlet and Two Exhaust Valves to each Cylinder

As, other things being equal, the weight of gas supplied per second by a pipe of diameter D varies as $D^{\frac{5}{2}}$, when using two smaller valves in place of one large one it must be remembered that on account of increased frictional losses the diameter of each of these must be $(\frac{1}{2})^{\frac{2}{5}} = 0·76$ at least, of that of the single large valve for equal supply of gas per second.

In the following table the diameter and lift of the inlet valves of 24 touring car engines of recent design are given; in each case the exhaust valve was of the same diameter as the inlet, and usually with the same lift; in a few instances the exhaust lift was from 0·03 in. to 0·06 in. greater than that of the inlet. The maximum practicable lift of petrol engine valves is about 0·5 in., partly from reasons of valve inertia and partly on account of the influence of the valve lift on the depth of the combustion chamber.

In the table the ratio of valve diameter to cylinder bore is given and also that of valve lift to diameter; it will be noted that practice has been by no means regular in these respects; the proportion of lift to diameter varies from 0·135 to 0·304, the average value of the ratio being 0·195, from which the bulk of the cases do not depart very far.

TABLE SHOWING VALVE PROPORTIONS FROM A NUMBER OF CASES OF PETROL CAR ENGINES, 1910

$d =$	$s =$	$\delta =$	$\lambda =$	$\frac{\delta}{d} =$	$\frac{\lambda}{\delta} =$
2·44	4·33	1·02	0·19	0·418	0·186
2·95	4·73	1·18	0·23	0·400	0·195
3·15	3·54	1·16	0·276	0·368	0·238
3·15	4·73	1·58	0·276	0·502	0·175
3·5	4·5	1·625	0·22	0·465	0·135
3·56	5·12	1·25	0·38	0·351	0·304
3·74	5·32	1·97	0·315	0·527	0·160
3·94	5·12	1·42	0·26	0·361	0·183
4·0	3·0	1·38	0·31	0·345	0·225
4·0	5·0	1·50	0·344	0·375	0·229
4·0	5·0	2·25	0·438	0·563	0·195
4·25	5·0	1·75	0·344	0·412	0·197
4·25	5·5	2·0	0·28	0·470	0·140
4·5	4·5	2·0	0·35	0·444	0·175
4·5	4·75	1·75	0·375	0·389	0·214
4·53	5·12	1·69	0·375	0·373	0·222
4·73	5·12	1·61	0·433	0·340	0·269
4·73	5·12	1·65	0·276	0·349	0·167
4·75	5·0	2·0	0·31	0·348	0·155
4·75	5·52	1·73	0·34	0·366	0·197
5·0	5·12	2·36	0·433	0·472	0·184
5·12	5·52	2·36	0·473	0·427	0·200
5·0	5·25	2·0	0·313	0·400	0·156
5·92	7·08	2·13	0·394	0·360	0·185

It is probable that a principal cause of the divergent power results so frequently obtained from tests of engines of the same size is this want of uniformity in valve and piping practice. In some cases valves have been purposely made small in order to prevent overrunning of the engine, and in these, of course, the power falls away rapidly with increase of speed beyond the normal rate. As mentioned above, the exhaust lift is occasionally a little greater than the inlet; it is doubtful if this is desirable if the exhaust valve periphery be all effective, for four reasons:

(1) That from inertia considerations the lift should be kept small.

(2) That as hot gas flows more freely than cold, a lift sufficient for the inlet should at least be sufficient for the exhaust.

(3) That there is usually a greater pressure available to cause flow of the exhaust gas than of the inlet; and

(4) That it is in general desirable to so proportion the exhaust valve outlet and piping as to obtain the best scavenging effect in the cylinder from the momentum of the exhaust gases in the exhaust pipe.

In connection with (4), an experiment by Mr. Poppe is of interest:

A series of tests was made with a four-cylinder, 80 mm. × 90 mm. engine under load supplied by an air brake dynamometer, so that the power varied as the cube of the speed, and with exhaust pipes of various diameters; the following results were obtained:

MR. POPPE'S EXPERIMENTS ON INFLUENCE OF EXHAUST PIPE DIAMETER

Dia. of exhaust pipe in ins.	Revs. per minute.	BHP
$1\frac{1}{8}$	1700	19·1
$\frac{27}{32}$	1720	19·8
$\frac{3}{4}$	1730	20·1
$\frac{5}{8}$	1700	19·1
$\frac{1}{2}$	1640	17·1
$\frac{13}{32}$	1470	12·0

Thus the maximum power was obtained with a $\frac{3}{4}$-in. pipe, and not with the largest pipe; this was attributed to the scavenging effect, due to the momentum of the escaping gas, being greatest in this case.

During the suction stroke $\frac{\pi}{4} d^2 s$ cub. in. of mixture enter the cylinder through the inlet valve. If the engine makes n revolutions per min. there are $\frac{n}{2}$ suction strokes per min., and hence $\frac{n}{2} . \frac{\pi}{4} . d^2 s$ cub. ins. of mixture pass the inlet valve in fifteen seconds of each minute.

If v'' denote the average velocity through the valve area $\frac{\pi}{4}\delta^2$, in ins. per min., we have therefore:

$$\frac{\pi}{4}\delta^2 v'' = 4\left\{\frac{n}{2}\cdot\frac{\pi}{4}\cdot d^2 s\right\}$$

$$= 3\pi d^2\frac{ns}{6}$$

$$= 3\pi d^2\sigma$$

Whence, if v denote the mean velocity in *feet* per min., as $v'' = 12\,v$, we get:

$$v = \left(\frac{d}{\delta}\right)^3 \sigma \tag{31}$$

where σ is the piston speed in feet per min.

Values of v, calculated in this way for nearly 150 engines in 1910,

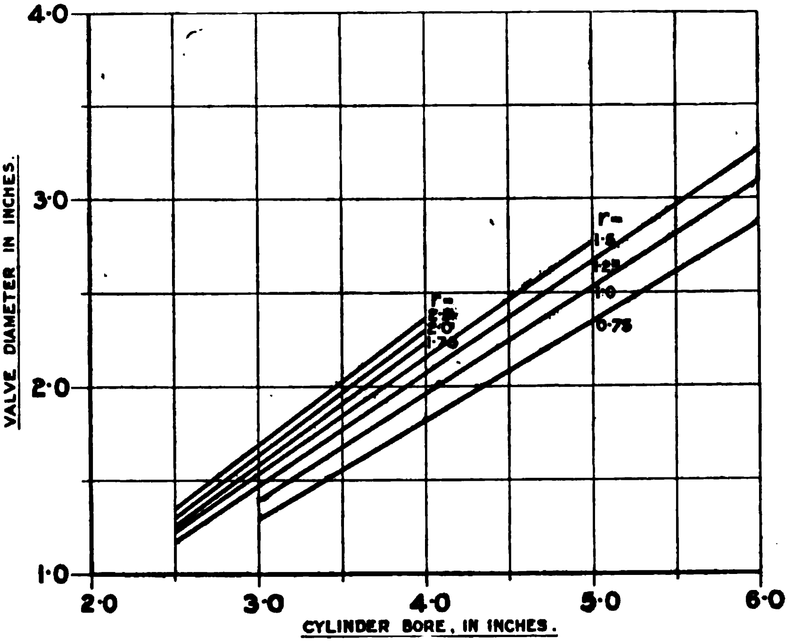


FIG. 316

showed that an average value of from 7000 to 8000 ft. per min. could be used in design[1]; hence for attainment of the best power results this should not be exceeded at the maximum piston speed. Substitute, then, in Eq. (31) 8000, say, for *v*, and for σ the maximum value as given in Eq. (11) *ante*, and we get on reduction :

---

[1] In more recent designs v is usually between 6500 and 7200 f.p.m. only.

$$\delta = 0{\cdot}287d\sqrt[4]{\frac{d^2s}{m}}\ \text{ins.} \qquad (32)$$

This equation connects the valve diameter with the cylinder bore, on the formula for maximum power suggested by Burls ; it furnishes regular and reasonable values, as will be seen from examination of the table given below, and curves in fig. 316. In the table, values of the lift are also given, obtained from the relation :

$$\lambda = 0{\cdot}195\ \delta\ \text{ins.} \qquad (33)$$

TABLE OF VALVE DIAMETER AND LIFT FROM EQS. (32) AND (33)

r =	d = Inches	δ = Inches	λ = Inches
0·75	3·0	1·29	0·252
	3·0	1·82	0·354
	5·0	2·38	0·463
	6·0	2·87	0·560
1·0	3·0	1·39	0·271
	4·0	1·98	0·386
	5·0	2·57	0·500
	6·0	3·07	0·600
1·25	2·5	1·16	0·226
	3·0	1·48	0·288
	4·0	2·08	0·405
	5·0	2·67	0·520
	6·0	3·26	0·635

r =	d = Inches	δ = Inches	λ = Inches
1·5	2·5	1·21	0·236
	3·0	1·52	0·296
	4·0	2·18	0·425
	5·0	2·77	0·540
1·75	2·5	1·25	0·244
	3·0	1·58	0·308
	4·0	2·22	0·433
2·0	2·5	1·28	0·250
	3·0	1·62	0·316
	4·0	2·28	0·445
2·5	2·5	1·36	0·265
	3·0	1·68	0·328
	4·0	2·37	0·462

The values of the lift are somewhat high for bores of 5 ins. and upwards ; it is probable that a smaller lift would suffice in these cases, however, on account of the rapid diminution of surface friction with increase in valve diameter. Some engine makers retain the valves of unchanged diameter and lift in different engines of a series for reasons of economy in production, but this practice cannot be defended if power output be a consideration of prime importance. In illustration of this, and also of the want of regularity frequently prevailing in valve sizes, the following figures are given for a series of engines of regularly increasing size by the same maker ; the upper line shows the actual inlet valve diameters, the lower the values obtained by aid of Eq. (32):

δ (actual) . . .	1·16	1·58	1·58	1·61	1·61	1·61	2·36
δ (calculated) .	1·31	1·40	1·63	1·63	1·80	1·97	2·08

The ratio of valve diameter to cylinder bore for the range of sizes usual in car engine practice, deduced from the table of p. 535 does not vary very far in value from 0·5, as the following list shows:

Ratios of $\frac{\delta}{d}$ from Table of Page 535

d. in ins.	Values of $\frac{\delta}{d}$ for r						
	0·75	1·0	1·25	1·5	1·75	2·0	2·5
2·5	—	—	0·464	0·484	0·500	0·512	0·544
3·0	0·430	0·463	0·493	0·507	0·527	0·540	0·560
4·0	0·455	0·495	0·520	0·545	0·555	0·570	0·593
5·0	0·476	0·512	0·534	0·553	—	—	—
6·0	0·478	0·513	0·543	—	—	—	—
Average	0·460	0·496	0·511	0·522	0·527	0·541	0·566

Mean value for the whole range = 0·518.

Hence, roundly, $\delta = \frac{1}{2}d$.

Now the Rating Rule of the Marine Motor Association (*vide* Eq. (30)) is:

$$\text{Rating per cyl.} = 3{\cdot}25 \times \frac{\pi}{4}\delta^2$$

whence, putting $\frac{1}{2}d$ for δ, we have on reduction:

$$\text{Rating per cyl.} = 0{\cdot}64d^2 \qquad (34)$$

so that in cases where the valves are proportioned in accordance with Eq. (32), the M.M.A. Rule will give results of the same type as those furnished by the R.A.C. formula, but about 60 per cent. higher.

Another mode of deducing a formula giving value diameter in terms of bore and stroke will now be given:

It is shown in Chap. IX, on 'Carburettors,' in this volume, that about 2·2 cub. ft. of mixture at atmospheric pressure and 60° F. are required per BHP per min.

Taking the rating recommendation of the Horse-Power Formula Joint Committee (*v.* Report, February 1911, and p. 526), we have:

$$\text{BHP per cyl.} = 0{\cdot}45\,(d+s)\,(d-1{\cdot}18) \qquad (22\text{A})$$

Hence each cylinder requires 2·2 times this, i.e. roundly

$$(d+s)\,(d-1{\cdot}18)\ \text{cub. ft. per min.}$$

Allowing a mean velocity of flow through the throat area of the inlet valve of 7000 ft. per min., and remembering that the mixture is only entering the cylinder during about one-quarter of each minute, we have as the necessary cross-sectional area of the valve throat in sq. ins.:

$$\frac{4 \times 144}{7000} \cdot (d + s)\ (d - 1\cdot18)$$

Denoting the throat diameter in inches by δ, we have, therefore, on equating and reducing:

$$\delta = 0\cdot324 \sqrt{(d + s)\ (d - 1\cdot18)} \text{ ins.} \qquad (32\text{A})$$

This result may be useful for purposes of design ; the following figures furnished by it serve to show that the sizes which result are in reasonable agreement with current practice:

d	=	2½″	3″	4″	5″
s	=	5″	5″	5″	5″
δ	=	1·02	1·24	1·63	2·00

Valve Spring Compression.—Notwithstanding the small mass of the valves and associated pieces, the high revolution speed of the small petrol engine necessitates the employment of extremely stiff helical springs in order that the valves may be caused to close with sufficient quickness. If the engine make n revolutions per minute, and if it be assumed that the time available for closing the valve is that occupied by the crank in turning through one-quarter of a revolution, then the time of closing will be expressed by $t = \dfrac{60}{n \times 4} = \dfrac{15}{n}$ seconds.

Now let m denote the mass of the valve, tappet rod, roller, &c., all complete, in lbs.

Let μ be the mass of the spring in lbs., and L its lift in feet. And let f denote the average compression-force in the spring, in lbs. weight. Then this average force of f lbs. weight has to accelerate m and $\frac{1}{2}\mu$; the average acceleration is thus $a = \dfrac{fg}{m + \frac{1}{2}\mu}$ f.s.s. But t being the time in seconds occupied in describing L ft. under acceleration 'a' f.s.s., we have $\text{L} = \frac{1}{2}at^2$ Hence, for this case:

$$\text{L} = \frac{1}{2}\left(\frac{fg}{m + \frac{1}{2}\mu}\right) t^2$$

Putting λ for the lift in ins., 32·2 for g, and $\dfrac{15}{n}$ for t, transposing and simplifying, there results finally for the value of the necessary average force of compression of the spring, in lbs. weight:

$$f = 11\cdot5\ (2m + \mu)\ n^2 \lambda \times 10^{-6} \qquad (32\text{B})$$

When the valve is seated the force of compression is less than this, and when fully lifted, greater; the difference, or 'accumulation' of spring-force, is somewhat considerable in the valve springs used in petrol engines, especially when of the racing type. For example, referring to the four-cylinder Vauxhall engine described in Chap. VIII of this volume, we may take $m = \frac{3}{4}$ and $\mu = \frac{1}{2}$, say; also $\lambda = \frac{1}{2}$ in.; thus, at 2500 revs. per min. we have, by (Eq. 32B):

$$f = 72 \text{ lbs. average.}$$

Now the approximate dimensions of the valve spring in this case are:

Spring wire: No. 5 S.W.G. (= 0·212 in. dia.). Mean diameter of the 12 coils = $1\frac{1}{2}$ ins. Hence by aid of the usual formula[1] for the compression of helical springs, one finds that the compression under load f lbs. is, in this case, $\frac{f}{100}$ ins. The average compression is thus 0·72 in., so that when the valve is seated the spring compression is 0·47 in., and when fully lifted, 0·97 in. At 0·47 in. compression the force is 47 lbs., and at 0·97 in. compression, 97 lbs.; the 'accumulation-ratio' is thus $\frac{97}{47} = 2{\cdot}06$. As actually tested in this case the values were 36 lbs. and 108 lbs.—an accumulation-ratio of 3·0. This example makes it clear that even with very light valves the springs must be very stiff when the engine is required to run at high revolution speeds, and this necessity imposes practical limitations upon the size of the valve; as already mentioned, the remedy lies in duplicating them.

VALVE SETTING

In normal engines for touring cars the inlet valve usually opens and closes a little late; the exhaust opens early and closes a little late. An examination of 25 touring car engines in 1909 showed the average valve setting to be as follows:

Inlet:	Opened 13° late	Closed 21° late
Exhaust:	Opened 45° early	Closed 6° late

The accompanying diagram, fig. 317, illustrates the setting adopted in the standard four-cylinder, 40 H.P. engine of the Wolseley Company. I_1 and I_2 are the crank positions for inlet valve opening and closing respectively; E_1 and E_2 are the positions for opening and closing of the exhaust. The arrow indicates the direction of rotation of the crankshaft.

The setting is as follows:

Inlet:	Opens 11° late	Closes 19° late
Exhaust:	Opens 38° early	Closes 7° late

---

[1] See e.g. *Kempe's Year Book*, 1912, p. 590.

In the six-cylinder, 38 H.P. Lanchester engine the timing adopted is:

Inlet:	Opens 7° late	Closes 20° late
Exhaust:	Opens 40° early	Closes 7° late

The inlet valve should not open until the pressure of the exhaust gases remaining in the cylinder has fallen below that in the inlet pipe near the valve, in order to prevent regurgitation, which may set up gaseous vibrations and affect the quality and quantity of the entering fresh charge. In some experiments by Dr. Watson ('Cantor Lectures,' 1910, p. 9), the pressure in the inlet pipe near the valve was, in general, rather less than atmospheric, the minimum ranging from about 1¼ lbs. per sq. in. below atmosphere at 650 r.p.m., to about 2¼ lbs. per sq. in. below at 1200 r.p.m.; only at the lower speed did the pressure, owing to gaseous momentum, ever momentarily exceed that of the atmosphere, and then by a fraction of 1 lb. per sq. in. only.

In four-cylinder racing engines an 'overlapping' setting is sometimes adopted, i.e. the inlet commences to open before the exhaust has quite closed, so that for a few degrees of the crank revolution both valves are partly open. This is done with a view to assist scavenging, and contrariwise to augment the entering fresh charge by aid of the momentum of the escaping gases. For very high-speed running, as e.g., on the Brooklands track, with a free exhaust, such a setting may be beneficial, but it is a matter of trial to determine whether it is of advantage in any particular case.

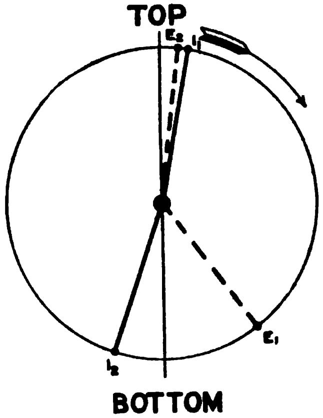


FIG. 317

An early opening of the exhaust valve tends to keep the engine cool, though at the expense of some loss of power and increase of exhaust noise. The best valve timing for any engine is dependent on the size of valves and arrangement of inlet and exhaust piping, and can only be finally settled by direct experiment; at high speeds the duration of valve opening has a marked effect upon the development of maximum power, and it is clearly an object in such cases to keep the valves open as long as possible. Such engines, however, will not usually run very satisfactorily at low speeds. Part of the advantage of the long-stroke engine arises from the possibility of setting the valves for a longer 'open' period than is practicable with a short stroke.

An instructive account of some experimental results obtained from a four-cylinder $4\frac{1}{2}$ ins. × 5 ins. engine having valves $1\frac{5}{8}$ ins. diameter with $\frac{5}{16}$ in. lift, at five different valve settings; appeared in the *Automotor Journal* for June 12, 1909. In the table on p. 541 the different valve settings are given, together with the maximum horse-power and corresponding speed obtained with each setting.

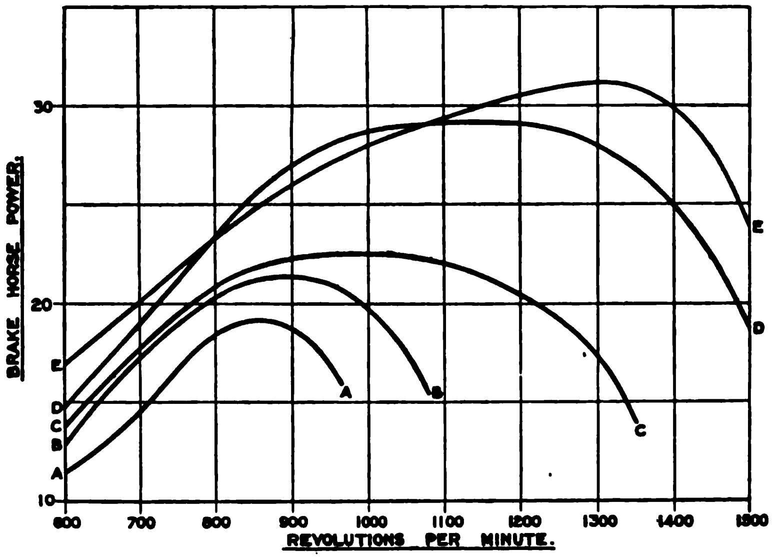


FIG. 318

In case A, the duration of opening of both valves was insufficient, and the passage of the gas into and away from the cylinder was obstructed; accordingly the maximum output was only 19·3 at 865 r.p.m.; the graph is shown in fig. 318. On increasing the duration of the 'open' period, and especially by earlier opening of the exhaust valve and later closing of the inlet, a progressive increase in speed and power resulted; thus in case E the BHP rises to 31·2 at a speed of 1300 r.p.m.

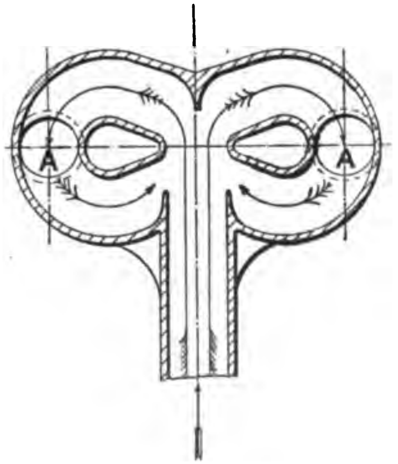


FIG. 319

D is a case of a 3° overlap setting, the inlet valve opening 9° late, while the exhaust valve does not close until 12° late; it will be seen that with this engine a better power and speed result was obtained with the non-overlapping setting E.

The accompanying fig. 318 shows the relation of power to speed with the five valve settings experimented with.

The inlet piping should be so arranged that each cylinder obtains its charge of working mixture without interference from the suction of any other cylinder. For two- or four-cylinder engines a figure of eight form, fig. 319, has been sometimes used with advantage.

Table showing Speeds and Maximum BHP obtained with Various Valve Settings. 4-cyl. 4½ ins. × 5 ins. Engine.

Ref. letter of test	Valve setting			Max. BHP obtained	Speed at Max. HP., r.p.m.
	—	Inlet	Exhaust		
A.	Opened	8° late	14° early	19·3	865
	Closed	0° late	0° late		
B.	Opened	8° late	30° early	21·5	900
	Closed	0° late	5° late		
C.	Opened	14° late	41° early	22·5	1000
	Closed	6° late	12° late		
D.	Opened	9° late	47° early	29·2	1150
	Closed	17° late	12° late (overlap)		
E.	Opened	13° late	39° early	31·2	1300
	Closed	22° late	10° late		

A, A are the inlet orifices to the cylinders. With the usual crank arrangement in two- and four-cylindered engines, the cylinders of one pair fire in consecutive strokes with a complete idle revolution following; in this case the circulatory flow of the gases in the figure-of-eight pipe, as indicated by the arrows, tends to minimise the abrupt changes in velocity which frequently occur with some other arrangements, and may thus help to equalise the charges to the two cylinders.

A common arrangement in four-cylindered engines is a four-branched inlet pipe, the four branches radiating from a short trunk pipe from the carburettor.

Many six-cylinder engines were at first prejudiced in their performance through an unsuitable arrangement of the inlet piping being adopted; it was often found that the first and sixth cylinders were 'starved'; insufficient attention also was given to the question of interference of one cylinder with another during the suction stroke, two inlets being open at the same time for a short interval with engines of this type.

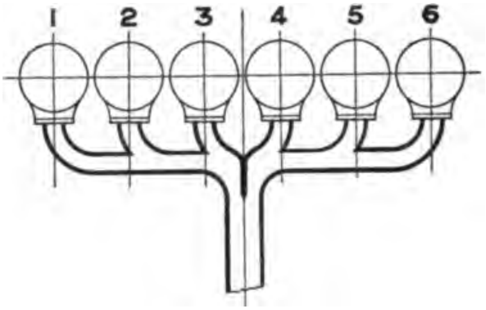


FIG. 320

In cases where the inlet orifices are all separate, Mr. Poppe adopts the arrangement shown diagrammatically in fig. 320; with the order of firing used, viz. 142635, it will be seen that no interference occurs during suction between any two of the cylinders.

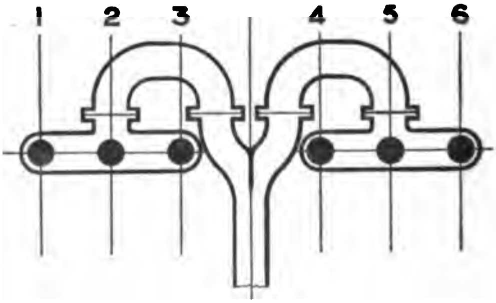


FIG. 321

An alternative arrangement of the inlet piping is shown in fig. 321, for six-cylinder engines with the firing order as before.

With reference to exhaust piping arrangements, it has already been pointed out that, to assist cooling, the exhaust gases should be carried as directly away from the cylinders as possible, and also that the influence of the exhaust pipe diameter upon the momentum of the escaping gases should be taken into consideration in design in order to obtain the best scavenging result.

In six-cylinder engines two cylinders are, for a short period, exhausting at the same time, and it is here important to so arrange that the exhaust from one cylinder shall be unable to blow back into another and thus impair its discharge. Such action may be prevented by connecting together only those cylinders that do not exhaust simultaneously; it will be seen that the piping arrangement shown in fig. 322 does this with the firing orders 142635, and 153624, two separate pipes being led to the silencer.

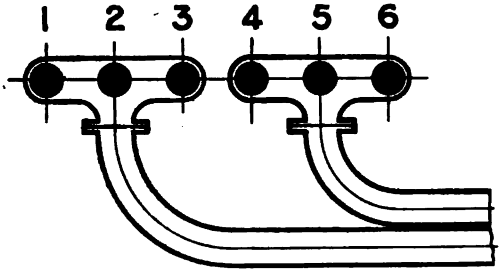


FIG 322

COMPRESSION

If V denote the volume of the combustion space in cub. ins., then the ratio $\dfrac{V + \dfrac{\pi}{4} d^2 s}{V}$ is termed the volume ratio of compression; if this be denoted by C, we have:

$$V = \frac{\dfrac{\pi}{4} d^2 s}{C - 1} \qquad (35)$$

The volume ratio of compression in car engine practice varies quite irregularly over a range of from 3 to 5; in the greater proportion of cases, however, the value lies between the narrower limits of $3\frac{1}{2}$ to $4\frac{1}{2}$. The table on p. 544 compiled from a number of recent engines of good design illustrates this; for the cases included the average value of C is 4·3, and this is a good general average. For the approximate estimation of compression pressure, designers commonly use the equation $PV^n =$ constant, where for n is taken the value $\frac{4}{3}$, TABLE SHOWING THE IRREGULAR VARIATION IN VALUE OF VOLUME RATIO OF COMPRESSION IN PETROL ENGINES FOR CARS; 1910 PRACTICE

Bore in inches	Stroke in inches	Volume compression ratio	Absolute pressure, lbs. per sq. in., by calculation
2·44	4·33	4·1	96
2·56	4·33	4·5	109
2·60	3·94	4·5	109
2·95	4·73	4·5	109
3·13	4·75	3·1	67
3·15	4·73	4·0	93
3·38	4·0	4·8	119
3·5	4·5	4·7	115
3·54	4·73	3·23	70
3·54	4·73	4·8	119
3·56	5·12	5·1	130
3·74	5·32	4·0	93
4·0	4·92	5·0	126
4·0	5·5	3·0	64
4·0	3·0	4·2	99
4·13	5·0	4·9	123
4·5	5·0	3·78	87
4·75	5·0	4·7	115
4·88	5·12	4·8	119
4·92	5·92	3·8	88
5·0	5·25	5·1	130
5·12	5·52	4·25	101
5·92	7·1	4·2	99

which is roughly its value for the mixture if adiabatically compressed. The constant is determined from the consideration that the initial volume at the end of the suction stroke is $V + \frac{\pi}{4} d^2 s$ at (app.) atmospheric pressure of 14·7 lbs. per sq. in.; the final volume is V, and if P be the corresponding compression pressure in lbs. per sq. in. absolute, then:

$$P V^n = 14{\cdot}7 \left(V + \frac{\pi}{4} d^2 s\right)^n$$

Thus:

$$P = 14{\cdot}7 \left(\frac{V + \frac{\pi}{4} d^2 s}{V}\right)^n$$

i.e. $P = 14{\cdot}7\, C^n$ lbs. per sq. in. abs. (36)

Putting $\frac{4}{3}$ for n, P is now readily obtained from the relation:

$$\text{Log } P = 1{\cdot}167317 + \frac{4}{3} \text{Log } C \qquad (37)$$

For convenience of reference some results from this equation are given hereunder:

c =	3·0	3·25	3·5	3·75	4·0	4·25	4·5	4·75	5·0
P =	64	71	78	86	93	101	109	117	126

lbs. per sq. in. abs. (app.).

The actual compression pressure attained depends in practice not only on the value of c, but also on the 'volumetric efficiency,' or proportion of actual to full charge obtained by the engine at various valve settings and different speeds; it may be determined with a limit of error of about 5 per cent. by use of the 'Okill' gauge or other specially designed compressometer.

Thus a 5 in. × 5½ in. engine at 2000 r.p.m. with c = 4·2 showed, by an Okill gauge, an actual compression of 99 lbs. per sq. in. abs.; with a Negretti & Zambra instrument on the same engine a compression pressure of 105 lbs. per sq. in. abs. was recorded. Again, in a 4 in. × 4 in. engine at 1400 r.p.m., with c = 4·7, the Okill gauge recorded a compression of 110 lbs. per sq. in. abs. These results are consistent.

In practice the limit of compression for touring car engines is reached at a value of c of about 5½, corresponding to a pressure of about 135 lbs. per sq. in. abs.; above this it is difficult to avoid pre-ignition; with kerosene even lower compression ratios must be used than with petrol.

In racing engines, required to run only for short periods and under special circumstances, higher compressions are sometimes employed.

Even with the moderate values usual in touring car engines the gradual increase in thickness of the carbonaceous deposit on the cylinder head and piston necessitates its periodic removal if pre-ignition is to be completely avoided; this trouble usually appears when the deposit attains a thickness of from 0·07 in. to 0·1 in. It can be delayed by polishing the top of the piston and the surface of the combustion chamber.

Theory indicates that, other things being equal, increased efficiency should occur with increased compression-ratio, and the air standard of comparison recommended by the Committee of the Inst.C.E. in 1906 implicitly involves this principle (*v.* Vol. I, pp. 248 *et seq.*). It was considered that the efficiencies of internal combustion engines of equally good design should be to one another in the same proportion as the corresponding efficiencies of air engines working on the same cycle and with the same compression ratios.

Apparent disagreements with this theory seem to result from many petrol engine tests, the thermal efficiency observed frequently appearing to be almost unaffected by considerable changes in the value of the compression ratio. Prof. Callendar has, however, shown very clearly that in such cases the theoretical gain due to increased compression-ratio does occur, but that this gain is in many cases partly, or even wholly, nullified owing to increased heat loss due to a greater ratio of surface to volume in the combustion chamber accompanying an increase in the compression ratio.

Prof. Callendar illustrates this by taking the figures from Dr. Watson's experiments (*v. Proc. Inst. A.E.*, III, pp. 457 *et seq.*); the following results were obtained at 700 r.p.m., and with an air-petrol mixture ratio of 14·5 to 1:

Compression ratio . .	3·92	4·35	4·71
Obs. thermal eff. . .	0·220	0·225	0·225
Air standard eff. . .	0·421	0·446	0·462
Obs. relative eff. . .	0·522	0·504	0·487

Thus the observed relative efficiency, instead of being constant in value, diminishes markedly with increasing compression ratio.

Now in these experiments the compression ratio was varied by the insertion of packing pieces between the foot of the cylinder and the crank-chamber, with the result that as the compression ratio was diminished, the ratio of surface to volume in the combustion chamber was also diminished as shown by the following figures:

Compression ratio . .	3·92	4·35	4·71
Ratio: $\frac{\text{Surface}}{\text{Volume}}$ . .	1·69	1·86	2·01

Hence it is at once clear that some of the advantage of the higher compression ratios must have been lost on account of increased heat loss due to the increased value of the surface-volume ratio; and Prof. Callendar shows that in this particular engine the increase in heat loss just sufficed to extinguish the compression-ratio gain.

The author has shown (Vol. I, p. 274) that the ideal efficiency, without any heat loss, of an actual internal combustion engine is about 0·8 of the corresponding air standard efficiency; taking the middle case, therefore, the ideal efficiency due to a compression ratio of 4·35 is $0·8 \times 0·446 = 0·357$. Hence the true relative efficiency in this case is $\frac{0·225}{0·357} = 0·63$; consequently the balance, viz. 0·37, of the theoretically utilisable heat has been lost by cooling. Now Prof. Callendar considers the cooling loss as proportional to the surface-volume ratio, whence the cooling loss in the first case will be $\frac{1·69}{1·86} \times 0·37 = 0·336$, and in the third, $\frac{2·01}{1·86} \times 0·37 = 0·40$; so that, on this view, the true relative efficiencies should be $(1 - 0·336)$, $(1 - 0·37)$, and $(1 - 0·40)$, i.e. 0·664, 0·63, and 0·60 respectively.

Accordingly the calculated relative efficiencies become 0·8 × 0·664, 0·8 × 0·63 and 0·8 × 0·60; that is:

Calculated rel. eff.	0·531	0·504	0·480
Observed rel. eff.	0·522	0·504	0·487

The calculated and observed relative efficiencies thus agree closely, justifying the assumption of proportionate heat loss and furnishing an explanation of the apparent divergence of the experimental results from the theory.

Prof. Callendar further points out that in order to attain the advantages of increased compression ratio on efficiency, this ratio should be increased by increasing the *stroke*, leaving the combustion chamber unaltered. And again, that the effect of increasing stroke but keeping the *compression ratio* constant must be to improve efficiency.

The deposit of badly conducting carbon which soon forms upon the piston head and combustion chamber walls diminishes heat loss and thus improves efficiency; it is probably due to this protective carbon coating that the diminished heat loss due to pocket walls permits these small engines to often show a very high efficiency notwithstanding the large value of the surface-volume ratio in the combustion chamber.

Moreover, it is not safe to conclude that by omitting pockets the efficiency necessarily increases on account of the diminished surface-volume ratio, as the surfaces in this case would probably remain cleaner and hence conduct better; there would, however, be less liability to pre-ignition.

The temperature of ignition for air-petrol mixtures is not yet definitely known; Prof. Hopkinson found from experiments with a Crossley coal-gas engine a pre-ignition temperature of about 1300° F.; Dr. Watson conjectures that for air-petrol it lies between 1100° F. and 1300° F.; it is, of course, pre-ignition that fixes the practical limit of the compression ratio of petrol engines between the values 5 and 6. On the other hand it must be borne in mind that increase of compression ratio is frequently, and sometimes necessarily, associated with increase in the surface-volume ratio, with consequent increase in proportionate heat loss by cooling, and hence it is probable that there is a compression ratio of maximum efficiency for each type of engine, which is attained when the efficiency gain due to raised compression ratio is just overtaken by the efficiency loss due to increased surface-volume ratio. Vol. I, pp. 303–4, may be referred to in this connection, where the report of the Gas Engine Research Committee is quoted to the effect that a series of experiments showed a rise of efficiency with compression from 0·28 to a maximum of 0·43, declining to 0·39 on the highest compression of all; and later: 'That in most cases the high efficiencies were associated with low percentages of heat lost in cooling water, thus supporting the view that the compression may be made too high in a given engine.'

High compression ratios, even when associated with increased efficiency, necessitate heavier construction and produce 'hard' running engines; for car work external considerations of comfort in riding accordingly have their influence upon engine design. Some makers deliberately use a low compression pressure in order to obtain soft and quiet running. The state of current practice in car engines can be inferred from the table given on p. 544, an average value of the compression ratio being roundly 4½.

For racing engines high compression ratios are usual; in addition to any gain resulting from thermal theory, there are the incidental advantages of increased suction during the in-take stroke, diminished dilution of the fresh mixture by the residue of the previous exhaust, and a greater practicable rapidity of ignition.

LUBRICATION

Mr. Morcom[1] has recently dealt very ably with the question of lubrication, and his valuable contribution to the subject is deserving of very careful consideration. He points out that the fundamental conclusion of modern theory—based largely on the experiments of Mr. Beauchamp Tower and the deductions from these of Prof. O. Reynolds—is that of the existence of a continuous film of oil between the rubbing surfaces of a properly lubricated bearing. The frictional resistance experienced arises from the shearing of this film, and may amount to as little as 1 per cent. of that of the same surfaces without lubrication.

The oil film thus prevents actual metallic contact with consequent risk of abrasion or 'seizing.'

The film pressure varies considerably at different points of a bearing, and the oil must be introduced at a point where the supply pressure exceeds the pressure in the film. The highest temperature occurs where the film has the least thickness, and continuous rotation with adequate oil supply maintains the continuity of the film.

Automatic lubrication occurs in a bearing subjected to an alternating load, due to the suction effect on the parting of the surfaces at each reversal of pressure, and heavier mean loads can be borne in such cases.

Failure may occur from an inadequate supply of lubricant; from local or general failure of the film arising from thinning of the oil through heat; or from excessive pressure crushing the oil film.

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[1] *Proc. Inst. A.E.*, May 1910.

The presence of grit and water in the oil reduces the stability of the film, and is, of course, to be avoided.

Mr. Morcom considers that forced lubrication to all bearings appears best to meet the rational requirements, and this system is very largely used, although of late many makers are fitting their engines with the 'semi-splash' system as described later.

For the lubrication of the petrol engines of cars mineral oils of high flash-point and well maintained high viscosity are most suitable.

The following table, compiled from data given by Mr. Morcom, illustrates the range of quality in the oils employed:

Flash-point by open test. °F	Viscosity by Redwood viscometer. Secs. for 50 c.c.	Car-miles per gallon	Notes
213	368 at 60 °F.	—	—
250	—	200	*Vide* Ch. VI, fig. 284
335	$\frac{\text{at } 70^\circ \text{ F.}}{\text{at } 212^\circ \text{ F.}} = 25$	—	—
338	—	1000	—
402	980 at 60° F. 39 ,, 250° F.	250	—
420	175 ,, 140° F.	600–1400	—
420	55 ,, 180° F.	1000	—
470	*50 ,, 210° F.	1130	*By Saybolt's viscometer; *v.* e.g. Redwood's 'Petroleum and its Products,' Vol. II, p. 604.
500	1130 ,, 120° F.	1000	
520	75 ,, 210° F.	700	
620	—	—	

The figures in the column 'Car-miles per gallon' depend, perhaps, a good deal on the personal element; they are very variable. A suitable flash-point is from 300° F. to 500° F. by open test.

The object of lubrication, then, is to prevent metallic contact occurring between relatively moving parts, and bearings are so proportioned that the intensity of pressure never becomes great enough to crush the oil film between the surfaces.

In ordinary touring car engines the connecting-rod and crankshaft bearings may be left easier than in cases where a very high speed is required; with very high speeds the pressure on the bearings, particularly during the later portions of the working stroke, becomes very great, due to the inertia of the reciprocating parts, and in order to resist this pressure either larger bearing areas or forced lubrication of the bearing must be provided. Further, at very high speeds the reversals of pressure at the crank-pin would cause knocking and vibration unless the bearings were fitted very closely, and hence again pressure lubrication becomes necessary for high speed work.

The various systems in use may be roughly grouped as follows:

(1) Splash lubrication.
(2) Semi-splash lubrication.
(3) Forced lubrication.

(1) *Splash Lubrication*, in which a mass of oil contained in the lower part of the crank chamber is maintained at an approximately constant level by means of occasional supplies from a hand pump connected with a reservoir on the dashboard, was in common use for car engines during several years, and is still employed on some of the smaller cars.

The connecting-rod end dips just beneath the surface at each revolution, and splashes the oil over the inside of the crank-chamber and to the piston and gudgeon; thence it trickles back, and finds its way into the crankshaft bearings through suitably provided oil-holes. The 'big-end' bearing is itself supplied with oil by a scoop-ended short pipe which dips into the oil at each revolution.

This simple method proves very satisfactory, especially with the smaller engines; with experience and a little care it is easy to supply just sufficient oil to provide practically effective lubrication without causing a smoky exhaust.

In some cases the oil reservoir on the dashboard is connected with the exhaust pipe, so that the pressure of the exhaust shall operate an adjustable sight-feed drip lubricator delivering into the crank-chamber; as the engine speed increases the exhaust pressure becomes greater, and the rate of drip consequently increases. In practice, however, this device is troublesome on account of the great variations in viscosity of the heavy mineral oils used for engine lubrication necessitating continual adjustment of the drip feed; in cold weather many of the lubricants used are semi-solid, and the drip device may then fail completely.

A better method is to employ a simple form of pump to either maintain a pressure in the reservoir or preserve a continuous circulation of the oil between reservoir and engine.

An objection to splash lubrication arises when an engine is inclined, as, for example, in a long ascent. In such a case the oil gravitates to the rearward end of the crank-chamber, so that the front end bearings are in danger of becoming short of lubricant and getting damaged. If the supply is maintained by a hand pump, and enters the engine at the front end, this risk is minimised.

(2) *Semi-Splash Lubrication.*—This system is now being increasingly adopted, particularly in English engines; an oil-pump, frequently of the gear-wheel type, draws oil from a sump formed in the bottom cover of the crank-chamber and delivers it to a trough under each 'big-end,' each trough being kept full to overflowing; the remaining

bearings are then lubricated by the big-end splash. In the sliding valve engines of the Daimler, Minerva, and Rover companies the troughs are raised and lowered automatically with the opening or closing of the throttle, so that the supply of oil is roughly proportioned to the speed of the engine. The overflow from the troughs and bearings gravitates back to the sump, usually through some simple filtering device.

Pockets and channels are so formed in the crank-chamber casting that the oil thrown up by the big-ends in running back down the interior surface of the chamber collects into them and is thus led to the main bearings, whence exuding it falls on to the filtering gauze screen at the bottom, and through this returns to the sump.

A refinement of the preceding method is to carry a considerable bulk of oil in a reservoir outside the engine instead of allowing it to collect in a sump ; the oil is cooled, and better lubrication is consequently obtained. Thus in the White & Poppe engines one oil pump supplies an excess of oil, for cooling as well as lubricating purposes, to each main bearing from an external reservoir. This oil next collects in troughs under the big-ends, and the overflow runs into a sump whence it is returned to the external reservoir by a second pump. The big-ends, gudgeons, and cylinder walls depend wholly on the 'splash' for their lubrication.

(3) *Forced Lubrication.*—On account of the reversals of pressure which necessarily take place at the crank-pin bearing in every complete cycle there is more need for pressure lubrication to the main bearings than to the big-ends. In high-speed racing engines, where the bearing pressures—as already pointed out—are very high, forced lubrication is practically a necessity, in order to keep the bearing area within practicable limits of size.

In the Mercedes-Knight sliding valve engine the oil is forced through a hollow crankshaft to the main bearings. The amount increases with the throttle opening, so that extra oil is supplied as the engine is speeded up.

In the 65 HP six-cylinder Napier engine the oil is forced by a pump to all the main bearings through suitable ducts drilled in the crankshaft.

In the 18–22 HP six-cylinder engine of the Sunbeam Co., a gear-pump, driven by skew wheels from the end of the exhaust-valve camshaft, delivers oil under considerable pressure to the crankshaft bearings, and thence by ducts drilled in the crank cheeks to the big-ends. The remaining bearings, &c., depend for their lubrication upon the splash from the big-ends. On its way from the pump to the main oil ducts the oil passes through a 'circulation indicator' and oil pressure regulator fixed on the dashboard.

Positive lubrication of the gudgeon bearing and cylinder walls is rarely met with ; it is a common experience that too much oil is thrown on to the under side of the pistons, and if this gets past the pistons a smoky exhaust is produced. This excess of oil comes mainly from the crank webs, whence it is whirled off, striking the pistons and cylinder walls with considerable velocity. In many pistons an oil-excluding spring ring, or 'scraper' ring, is fitted near the bottom, with the object of preventing this excess oil from getting into the combustion chamber ; an old device was to fit a baffle plate between the cylinder and crank-chamber, with a narrow slot for the connecting-rod to swing in, to intercept the oil thrown up by the crankshaft. In the Vauxhall engine a number of small holes have been drilled in the pistons below the rings, the theory being that the superfluous oil on the cylinder walls on reaching the holes is forced into them and thence falls back into the crank-chamber. Messrs. de Dion fit a small 'thrower' ring to the upper half of the main bearings which catches the oil whirled upwards from the crank webs and prevents it reaching the piston and cylinder walls ; this device is effective.

The gudgeon bearing is heavily loaded and has but a small motion ; experience shows that it is frequently here that the first slackness occurs in an engine. The oil thrown up from below is frequently partly carbonised by contact with the hot piston, and the gudgeon accordingly, with the usual practice, is likely to be served with a hot, thin, and gritty lubricant. With pressure feed to the gudgeon a supply of cool, clean oil is assured, and grit is excluded ; hence positive lubrication of the gudgeon is to be considered good practice if suitable means be provided for preventing excess of oil passing the pistons and reaching the combustion chamber. Thus in the Rolls-Royce and Lanchester engines the gudgeons are pressure fed. In the former case an oil pump draws from a sump and forces the oil into each main bearing and big-end ; a pipe clipped to the connecting-rod conveys oil from the big-end to the gudgeon. A supplementary oil reservoir is fitted to replenish the sump when necessary ; an oil regulating device working in unison with the throttle provides for increased lubrication at high speeds.

In Lanchester's engine also the lubrication is forced into all the main bearings, big-ends and gudgeons, returning by gravity to the base of the crank-chamber. The oil duct to the gudgeon is in this design formed in the connecting-rod itself, being drilled along a swelled bead in the centre of the web.

In this engine the cam-shafts are partly external, and the cam-shaft bearings are oiled by hand about once a month, the oil being contained in troughs through which the shafts pass.

The advantages of the forced lubrication system are : (1) smaller bearing surfaces may be used, and (2) the oil ducts do not become filled up.

Among the drawbacks are: (1) the big-end lubrication depends on the condition of the main bearings; to ensure sufficient oil when the main bearings are worn, an excess must be allowed when these are tight, with the result that too much is thrown off from the big-ends, causing sooting-up and smoky exhaust: (2) the large variations in the viscosity of the engine lubricating oil render the readings of the oil pressure gauge on the dashboard frequently misleading: and (3) the pressure system may break down. Few engines contain any provision for running on the 'splash' system in this event, though in many cases this could be arranged quite simply, and would be a valuable stand-by in case of need.

The oil pressure used varies considerably in practice, rising in some instances as high as 40 lbs. per sq. in.; it diminishes as the bearings become worn.

Mr. Morcom, from a large experience, uses the following empirical rule for determining the capacity of the oil pump: Delivery in cub. ins. per min. at maximum speed = 8 × Sum of the peripheries of all the bearings in ins. This agrees very well with current practice.

The nominal oil pressure in the De Dion engines is about 10 lbs. per sq. in.; in the Lanchester about 40 lbs. per sq. in. In some of their large petrol engines for marine work, running at a comparatively slow speed, Messrs. Thornycroft have successfully used the very high pressure of 100 lbs. per sq. in.

COOLING

The early car engines of Daimler and Panhard were water cooled, while the first small de Dions, Renaults, Argylls, New Orleans, &c., were air cooled; air cooling soon proved unsatisfactory in car work, and engines with few exceptions were water cooled. Messrs. Lanchester retained air cooling in their well-known design as late as 1907; this system is now only found in the engines of cycles and aeroplanes.

Circulation of the cooling water was at first maintained by reciprocating pumps; but these were soon replaced by the rotary types, usually either a simple centrifugal, or, more commonly, a pump of the gear-wheel or 'Root's blower' variety, fig. 323, fitted to run at a normal peripheral speed of about 1000 f.p.m. Circulating pumps were at first frequently driven by frictional contact with the rim of the engine flywheel, but are now always positively operated by gearing from the crankshaft.

Early cylinder designs were frequently defective in their cooling jacket details; a common source of trouble was the over-heating of the exhaust valve and spindle; 'pockets' were sometimes left in the jacket spaces wherein the cooling water could not enter or wherein there was little or no circulation, and the formation of steam in these, followed by the expulsion of the water and consequent over-heating of the cylinder, was a not unusual experience.

The water circulating pipes between the several cylinders, radiator, and pump were generally too small in diameter, and the pump in consequence was unnecessarily loaded. Cases of engines with very small water pipes are still not uncommon. Within the past two years, however, designers have more fully recognised the practical benefits of large connecting pipes and lower velocities of flow, and many engines now have water pipes with pump circulation so large and so directly arranged, that it is probable the action would be maintained, even should the pump fail, by the 'thermo-syphonic' action alone.

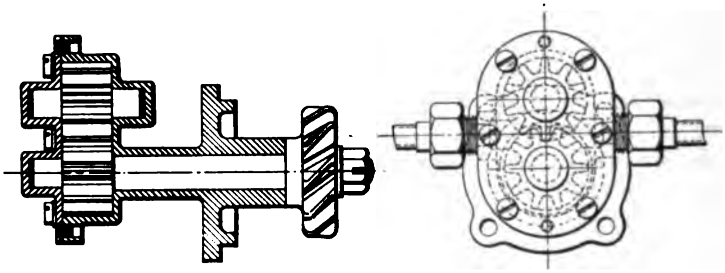

FIG. 323

The tendency of recent designs is, in fact, to omit the water pump, and so design the water system that the circulation is maintained by the difference in the density of the heated and cooled water only. This, termed the 'thermo-syphon' system of cooling, furnishes a very small head, and hence necessitates low velocities of flow, and consequently connecting pipes of larger diameter, with small frictional resistance in the radiator and cylinder passages. The system is particularly applicable to engines having their cylinders 'en bloc,' and in such cases adds to the simplicity and effectiveness of the design; with separately cast cylinders it is sometimes awkward to provide the large water connections necessary.

In fig. 324 a sectional view is given of the four-cylinder, 12–16 HP Wolseley engine with cylinders 'en bloc' and thermo-syphon cooling.

The radiator is usually placed in front of the engine; Messrs. Renault Frères, however, who employ thermo-syphon cooling in all their engines, fit the radiator on the front of the dashboard, thus giving free access to the engine and accessories when the bonnet is raised.

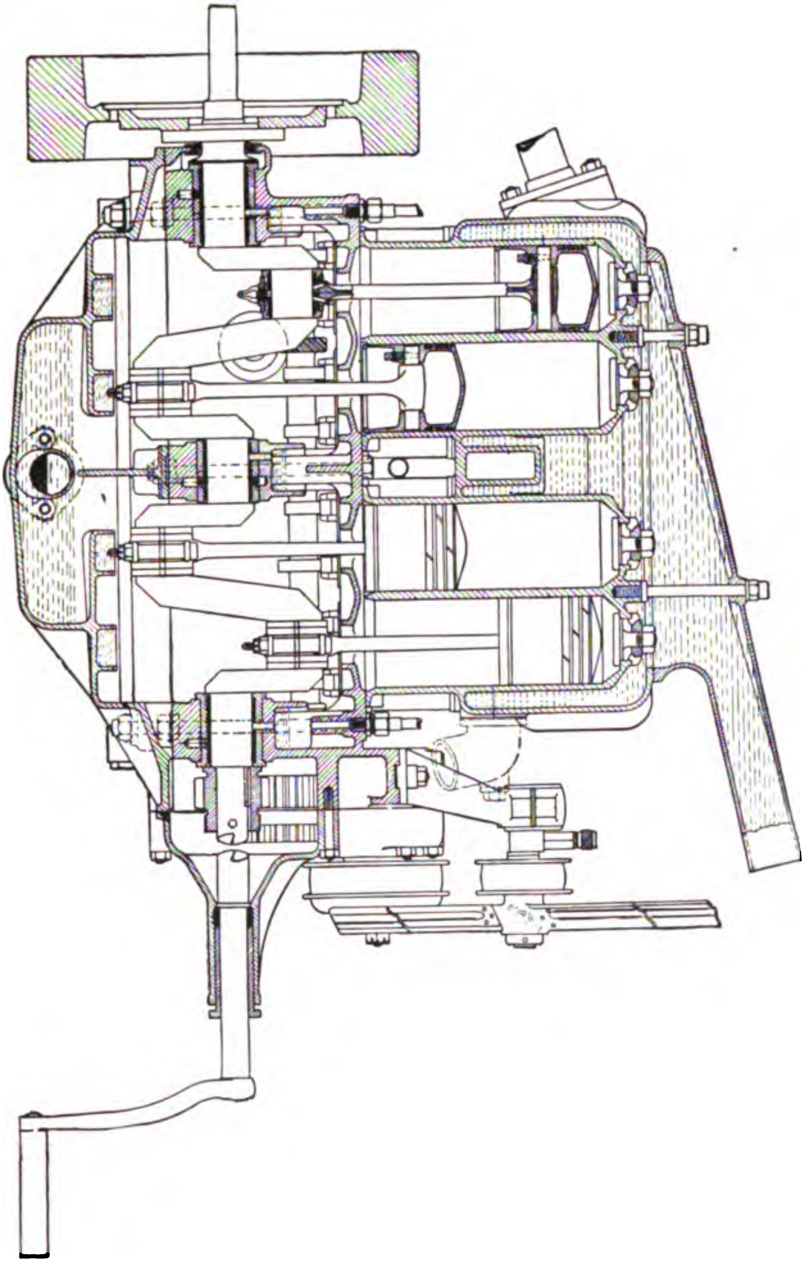

FIG. 324

The arrangement of the four-cylinder, 14–20 HP engine is shown diagrammatically in fig. 325.

The following short table illustrates the extent to which the thermo-syphon system of cooling is now adopted by builders of experience.

SOME 1911 CARS WITH THERMO-SYPHON COOLING

Name of engine	No. of cyls.	Cylinder arrangement	Bore and stroke, mm.	Nominal HP
Ariel	4	en bloc	70 × 100	12·1
Armstrong-Whitworth .	4	en bloc	$\left.\begin{matrix}80\\85\end{matrix}\right\} \times 120$	16 & 18
Crossley	4	en bloc	102 × 140	20
Delage	6	en bloc	66 × 125	—
Lorraine-Dietrich . .	4	en bloc	75 × 120	12–16
Metallurgique . . .	4	en bloc	75 × 110	12–14
Imperia	4	en bloc	75 × 100	12
Panhard.	4	en bloc	80 × 120	12–15
Sizaire	4	en bloc	70 × 120	12·1
Thornycroft . . .	4	en bloc	102 × 114	18
Vauxhall	4	en bloc	90 × 120	20
Wolseley	4	en bloc	79 × 114	12–16
Alldays	4	in pairs	86 × 108	14–18
Argyll	4	in pairs	80 × 120	15
Berliet	4	in pairs	100 × 140	20–25
Clément	4	in pairs	85 × 120	14–18
de Dion	4	in pairs	66 × 120	12
Deasy	4	in pairs	80 × 130	14–20
Gregoire	4	in pairs	80 × 160	16–22
Lanchester	6	separate	102 × 76	28
Napier	4	in pairs	82 × 127	15
Maudslay	4	in pairs	90 × 130	17
Renault	4	in pairs	All engines	—
Sheffield-Simplex . .	6	in pairs	85 × 127	25
Swift	2	one pair	102 × 111	10–12
Vulcan	4	in pairs	80–120	15·9
White & Poppe . .	4	in pairs	80 × 130	15·9
Humber	4	in pairs	—	12

The radiator consists of an upper reservoir, A, and a lower, B, connected by a 'forest' of smooth vertical copper pipes, C. The bonnet and under shield completely enclose the engine space; the flywheel has fan arms which, during running, draw the air from this space; fresh air consequently flows in through the exposed portions of the radiator on each side of the bonnet, and, passing across the surface of the copper pipes, cools the circulating water. The slope of the pipe connecting the engine with the upper reservoir should exceed that of the steepest hill the car may ever be required to climb. With thermo-syphon cooling it is essential to the action that the water level shall always be above the top of the rising pipe.

Mr. Poppe considers that in a new engine with thermo-syphon cooling the temperature of the water on leaving the cylinders should not exceed 175° F., and prefers the plain tube type of radiator, as offering little resistance to the passage of the water and not becoming so easily diminished in efficiency after prolonged service by the action of accumulated deposit.

In the most usual arrangement, with the radiator in front of the engine, an air fan band-driven from the engine crankshaft is placed close behind the radiator to maintain a rapid flow of air through it at all times. Messrs. Napier employ a very neat design of fan with eight vanes and stiffening rim, all pressed from one sheet of steel; the boss to which the fan is riveted is of aluminium cast under pressure, and is fitted with a pair of ball races lubricated by an external greaser.

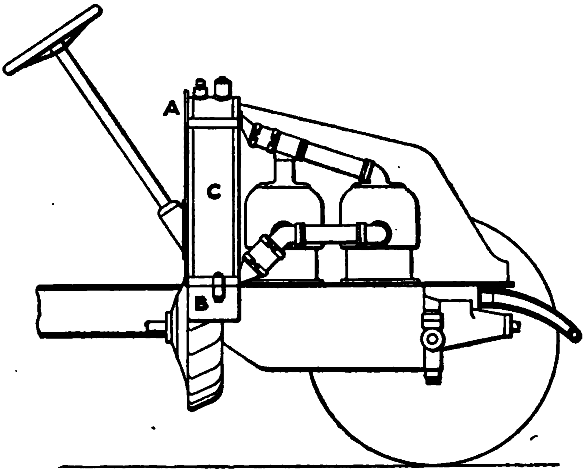

FIG. 325

In winter the cooling arrangements prove sometimes *too* effective, and the fan driving band may, in such event, be removed with benefit to the running of the engine.

ON TORQUE

If T denote the flywheel torque per cylinder, in lb.-feet, corresponding to HP brake horse-power at *n* revolutions per minute, then

$2\pi n T = 33{,}000$ HP; whence, putting for n its value $\frac{6\sigma}{s}$ we have on reduction:

$$T = 875 \frac{HP}{\sigma} . s \qquad (38)$$

where s is in inches, and σ in feet per minute.

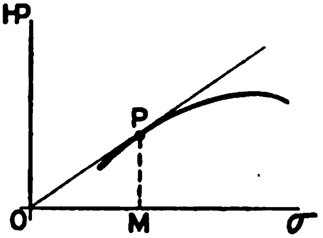

FIG. 326

Thus T is a maximum with the ratio $\frac{HP}{\sigma}$; hence (fig. 326) if on the HP σ graph a tangent be drawn to the curve from zero origin, the point of contact determines the piston speed and horse-power per cylinder at which the torque is a maximum.

Thus in fig. 326 the maximum torque occurs at piston speed OM, and horse-power per cylinder MP, and its value per cylinder is

$$T_{max.} = 875 \frac{MP}{OM} . s \text{ lb.-feet}$$

MP and OM being measured by their respective scales.

Some useful inter-relations may be here stated; thus Eq. (14) may be written:

$$\eta p = 168{,}000 .. \frac{1}{d^2} . \frac{HP}{\sigma}$$

and substituting for $\frac{HP}{\sigma}$ from (38) we obtain on reduction:

$$\eta p = 192 . \frac{T}{d^2 s} \text{ lbs. per sq. in.} \qquad (39)$$

Hence the brake mean effective pressure and the torque per cylinder are mutually proportional; the same graph, with change of scale only, will accordingly suffice to represent both the ηp and the T variations with speed.

Again, if W lbs. of petrol be supplied to the engine per cylinder per minute, the corresponding heat supply per cylinder-minute is roundly 18,600 × W B.Th.U. If τ denote the brake thermal efficiency, the work-heat per cylinder-minute is 18,600 × W × τ B.Th.U., and the BHP per cylinder is accordingly expressed by:

$$HP = \frac{778}{33{,}000} \times 18{,}600 . W . \tau$$

i.e.

$$HP = 438{\cdot}5 . W . \tau \qquad (39A)$$

Substitute now this value for HP in Eq. (38), and we get:

$$\text{T} = 383{,}688 \;.\; \frac{\text{W}\tau}{\sigma} \;.\; s \text{ lb.-feet per cylinder.} \qquad (39\text{B})$$

Thus torque varies directly as fuel supply and directly as the thermal efficiency.

As $\sigma = \frac{ns}{6}$, Eq. (39B) can be written in the simpler form:

$$\text{T} = 2{\cdot}3 \times \frac{\text{W}\tau}{n} \times 10^6 \text{ (app.)} \qquad (39\text{C})$$

Finally, substitute this value of T in Eq. (39), and we have on reduction:

$$\eta p = 4{\cdot}42 \;.\; \frac{\text{W}\tau}{d^2sn} \;.\; 10^8 \text{ (app.) lbs. per sq. in.} \qquad (39\text{D})$$

Hence, also, the brake mean effective pressure varies directly as the fuel supply, directly as the thermal efficiency, and inversely as the product d^2sn.

Until recently (1911) it was common in car engines to find the maximum torque occurring at the low piston speed of 400 or 500 ft. per min.; recent improvements in design have resulted in the production of engines wherein the torque varies but little over a wide range of speed, and attains a maximum value at a piston speed in some cases exceeding 1200 ft. per min.; instances will be found in the next chapter.

In his presidential address to the Institution of Automobile Engineers in 1908 Clerk discussed the question of the flexibility of petrol engines, and pointed out that ideal flexibility implies ability to develop maximum power at all speeds, so that the torque would vary inversely as the speed (Eq. (38)); such a property is quite impossible in a petrol engine working on the usual cycle; at best the torque can be kept at practically a constant value over a large range of speed, so that *at best* the petrol engine develops a power proportional to its speed, and thus falls far short of the ideal condition. At very low speeds the increased cooling loss, arising from the working charge remaining in the cylinder for a longer time, causes ηp, and consequently the torque, to fall off very rapidly.

Many attempts have been made to increase torque by supplying the working mixture to the engine under pressure. The proposal originated with Daimler, and has been tried also by Dawson, Clerk, O'Gorman, and others. The charge weight is thus increased and (*v.* Eq. (39C) in this way the torque may be considerably increased. The highest value of ηp in ordinary petrol engines is about 110 lbs. per sq.

in.; a value of ηp of 150 lbs. per sq. in. can be attained in this manner, though, as was seen in the case of the Dawson engine, at the cost of considerable increase in engine weight in order to withstand the larger forces created.

Clerk has made experiments on gas engines with various super-compression devices, and has found it possible to increase ηp at low speeds by about 50 per cent. *without increase of the explosion pressure*; this result was achieved by supplying the charge under pressure—as proposed by Daimler—and at the same time increasing the volume of the compression space. In this way the torque was greatly increased at slow speeds, although, of course, at the cost of a diminished compression ratio and some loss of theoretical efficiency.

Some torque curves from bench tests of recent engines are given in Chap. VIII of this volume, and may be referred to with advantage in this connection.

MECHANICAL EFFICIENCY, η

The mechanical efficiency is the ratio of the brake to the indicated horse-power. The mechanical *loss* is the difference between the indicated and brake powers. The mechanical loss in an engine working on the Otto cycle is made up of (*a*) the true frictional losses, due to pistons, valves, bearings, &c., and (*b*) the pumping losses due to the nature of the cycle.

The mechanical *loss* is found to vary greatly in internal combustion engines with variation in the jacket water temperature, and in the mode of lubrication adopted. Thus Prof. Hopkinson in his experiments on an 11·5 ins. × 21 ins. Crossley gas engine at about 180 r.p.m. with the exhaust valve cover removed so that there was no compression, obtained the following results:

With normal lubrication, an increase in jacket water temperature from 70° F. to 180° F. caused a diminution in the mechanical loss from 6½ to 4 HP.

With excessive lubrication the loss at 70° F. was reduced from 6½ HP to 4¾ HP, while by injecting water into the cylinder, notwithstanding the absence of compression, the loss at 70° F. was said to have been reduced to the low value of 2¾ HP.

The experiments of Mr. Morse with a four-cylinder, 3·56 ins. × 5·11 ins. Daimler engine are given in the *Proceedings of the Inst. A.E.*, Vol. III. The engine was fitted with a Prony brake, and was run at full load, the BHP being then noted. The ignition of one cylinder was then cut off, and the brake adjusted until the engine speed again attained its full load value; subject to certain reductions, the difference of the two BHP's was taken as a measure of the IHP of the non-firing cylinder. This process was repeated for each cylinder in succession, and the IHP of the complete engine thence inferred.

The following results were obtained with splash lubrication and jacket water temperature of 212° F.:

Revs. per min.	720	1000	1220
Brake HP	11·80	14·72	17·10
Friction HP	0·74	1·55	2·50
Pumping loss HP	0·38	0·64	1·42
Indicated HP	12·92	16·91	21·02
Mech. eff., η	0·914	0·871	0·814

These figures are at full throttle, and it will be noted that at all three speeds the pumping loss is roughly one-half of the engine friction loss, i.e. about one-third of the whole mechanical loss for this engine. When throttled down, however, the pumping losses become relatively more important, and cause the low mechanical efficiency of engines of this type at light loads, as the following experimental results show:

—	Full load	Half load	Light
Revs. per min.	720	720	720
Brake HP	14·5	7·4	0·6
Friction HP	0·74	0·74	0·74
Pumping loss HP	0·36	0·58	1·45
Indicated HP	15·6	8·72	2·79
Mech. eff., η	0·93	0·85	0·215

In the accompanying table some values of η are collected together for convenience of reference:

VALUES OF η FOR SOME PETROL ENGINES

Name of engine	No. of cyls.	Bore and stroke (inches)	Compression ratio	Jacket water tempt.	At — load	Revs. per min.	Mech. eff., η	Observer
Siddeley	4	4·62 ×5·08	4·18	Normal working	full	530	0·89	Hopkinson
					full	930	0·84	
Daimler 1906	4	3·56 ×5·11	3·85	212°F.	full	720	0·91	Morse
					full	1000	0·87	
					full	1220	0·81	
					full	720	0·93	
					half	720	0·85	
					4%	720	0·215	
Talbot .	4	3.34 ×4.72	4·71	Normal working	0.5	700	0·835	Watson
					0.8	5050	0·83	
					full	1285	0·82	
Clèment	1	2·36 × 2·74	3·75	Air cooled	full	1270	0·80	Callendar

Thus the mechanical efficiency diminishes with increase of speed and with reduction of engine load. It is lower when an engine is new than after it has been run for some time with load. Its value is affected by the jacket water temperature, and by the method and extent of lubrication. On the whole, for normal conditions of running with water-cooled petrol engines, an average value of about 0·85 can generally be safely taken in ordinary calculations.

Prof. Callendar (*Proc. Inst. A.E.*, III, p. 261), from an examination of the large L, R, & X gas engine trial results (*v.* Vol. I, pp. 250–6), in conjunction with experiments of his own upon the small air-cooled Clèment petrol engine mentioned in the preceding table, finds that for these four cases the variation of η with size is expressible in the form:

$$\eta = 0{\cdot}91\left(1 - \frac{0{\cdot}3}{d}\right) \qquad (39\text{E})$$

The following short table enables a comparison to be made between observed values of η and those furnished by this equation:

Engine	d, in inches	Value of η from		Notes
		Experiment	Eq. (39 E)	
Clèment . .	2·36	0·80	0·80	Callendar
Talbot . .	3·34	0·82	0·83	Watson
Daimler . .	3·56	0·87	0·83	Morse
Siddeley . .	4·62	0·84	0·85	Hopkinson
L	5·5	0·84	0·87	Inst. C.E.
R	9·0	0·85	0·88	*v.* Vol. I, p. 252
Crossley . .	11·5	0·86	0·89	Hopkinson
X	14·0	0·86	0·89	Inst. C.E.

With reference to engines L, R, and X, Vol. I, pp. 250–1, may be referred to; it is there shown that by one mode of estimation the values of η deduced from the trial results are 0·86, 0·87, and 0·89 respectively, which are in close agreement with the figures furnished by Eq. (39E).

But it is clear from what has already been stated as to the nature and extent of the variations in the value of η that no simple expression can possibly express its value accurately; in general considerations of power rating, however, Eq. (39E) is of interest and value.

A very full discussion of the mechanical efficiency of the Otto type of engine is given by the author in Vol. I, pp. 250–82; p. 278 may particularly be referred to here in connection with the analysis made of the mechanical loss in the case of the Crossley engine. It will be noted that the loss due to piston friction amounts to 6·1 per cent. of the indicated HP, and that it is equal to the sum of the pumping and other frictional losses.

Piston friction in this case thus accounts for 50 per cent. of the mechanical loss; this friction is, in general, large and variable, being dependent on the condition of fit of piston and rings, age of the engine, nature and extent of lubrication, and temperature of jacket water. It is also affected by the form of the head, i.e. whether domed, flat, or dished, inasmuch as the piston distortion under the working impulses varies with the type adopted.

THERMAL EFFICIENCY

Among the many valuable contributions of Dr. W. Watson, F.R.S., to the science of the petrol engine is a Paper on the thermal efficiency of a petrol motor presented to the Institution of Automobile Engineers in May 1909.

Prof. Watson's tests were conducted on a four-cylinder, 3·55 ins. × 4·73 ins. Clèment-Talbot engine; three sets of experiments, referred to as the A, B, & C series respectively, were made with different compression pressures.

It was found that owing to wire-drawing and exhaust back pressure the actual compression pressure attained diminished with increase of engine speed, as shown by the following figures:

Series	A	B	C
Vol. ratio of compression . . .	4·71	4·35	3·92
Calculated absolute pressure, Eq. (37).	116·0	104·0	90·8
Actual compression pressure, attained, lbs. per sq. in. abs., at 800 revs. .	102·0	94·0	84·7
Actual compression pressure attained, lbs. per sq. in. abs., at 1600 revs. .	94·7	86·7	76·7

The petrol used was Pratt's, having a specific gravity of 0·719 at 60° F., and a composition by weight of:

$$C = 0{\cdot}8522$$
$$H = 0{\cdot}1478,$$

proportions practically identical with those resulting from an earlier determination by Prof. B. Hopkinson; hence 1 lb. of this petrol requires for its complete combustion to CO_2 and H_2O 14·8 lbs. of air. The calorific value of the petrol was roundly 16,770 B.Th.U. per pint at 60° F. The exhaust gas analysis showed, however, that combustion was practically complete when only 14 lbs. of air were supplied per lb. of petrol, and hence Prof. Watson conjectures that the combustion of the petrol is not quite so simple as is customarily assumed; some experiments showed, for example, that aldehyde $(C_nH_{2n}O$ was present in the exhaust gases, and this may account for some of the anomalies observed.

The proportion by weight of air to petrol varied in the experiments from 10 to 20, roundly. It was found that the thermal efficiency, calculated in the usual way, increased as the mixture became weaker, up to very nearly the weakest mixture with which the engine would run regularly. This result is in agreement with that of the author in the case of gas engines, as first stated by him in 1882 (*v.* Vol. I, p. 283).

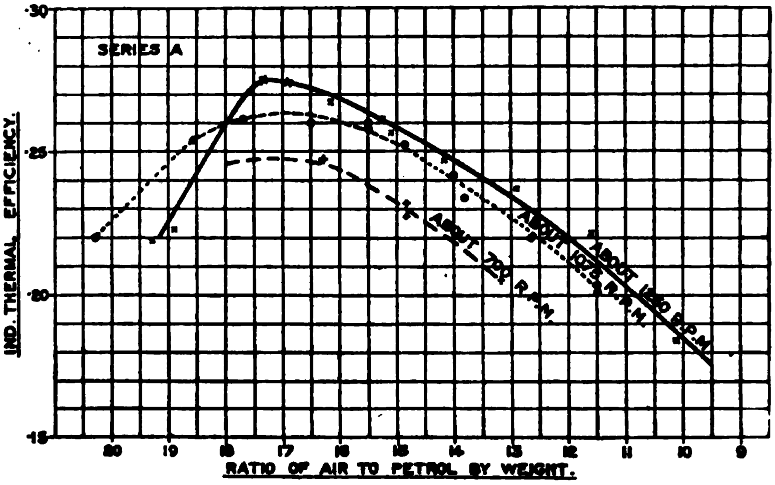

FIG. 327

In fig. 327 the results of Prof. Watson's tests are resumed; the thermal efficiency is plotted against the ratio of air to petrol by weight; the three curves given correspond to speeds of approximately 1250, 1075, and 700 r.p.m. respectively, for the series A.

It will be noted that at all three speeds the thermal efficiency is at a maximum value for a ratio of air to petrol of about 17½, and that the thermal efficiency increases with the speed.

Usually in estimating the thermal efficiency of a petrol engine it is assumed that the whole calorific value of the petrol used is liberated, which is probably never the case, and hence the efficiency figures obtained are too low. Prof. Watson determined the fraction of the total heat actually liberated in the engine from analyses of the exhaust gases in his tests, and his results, given in the table on p. 565, show that the fraction varies from 0·64 to 0·99; the corresponding corrections in the figures for thermal efficiency are also given:

SERIES A

Ratio of air to petrol by weight	Fraction of heat of fuel liberated	Thermal efficiency calculated from	
		Whole calorific value of fuel used	Fraction actually liberated
10	0·64	0·185	0·289
11	0·71	0·204	0·287
12	0·79	0·220	0·278
13	0·89	0·235	0·264
14	0·99	0·248	0·251

Hence the thermal efficiency, estimated in this way, *increases* with the richness of the mixture used by the engine.

Another consideration to be borne in mind in comparing the thermal efficiency with the corresponding air standard efficiency is that the actual working substance is *not* ideal air with a constant specific heat, but a mixture of gases having an apparent specific heat which increases considerably with the temperature. The properties of the mixture of working gases in a gas engine have been investigated by the author, and his results are given in Vol. I, p. 235; on pp. 267–75 of the same volume will be found the application of these results to the estimation of the maximum possible thermal efficiency, for a given compression ratio, of an engine using this mixture of gases. Broadly, the conclusion reached is that the 'air standard' furnishes figures unattainably high, the true maxima values being roundly 20 per cent. less than the air standard values indicate.

In the petrol engine the mixture of gases after ignition is generally similar to that in a gas engine, and hence this conclusion may be applied here also. In the case of the petrol engine, however, the change of volume of the mixture on combustion is variable and considerable, being an *increase* of about 6 per cent. for an air-petrol ratio of 14, and 14 per cent. when the ratio is reduced to 10; this differs markedly from cases where coal gas is employed, a *contraction* of about 3 per cent. occurring with ordinary mixtures (*v.* Vol I, pp. 330–1).

Considering, then, that the actual maximum efficiency possible with a given compression ratio is not that of the air standard, but roundly 20 per cent. less, we obtain the following as the values of the relative efficiencies from Dr. Watson's tests, the figures referring to the mixture giving the highest thermal efficiency, viz. about 17½ for the ratio by weight of air to petrol.

Series	Compression ratio	Air standard efficiency	Maximum efficiency possible.	Actual thermal efficiency	Relative efficiency	
					Compared with air standard	Compared with max. possible
A	4·71	0·46	0·37	0·28	0·60	0·76
B	4·35	0·44	0·35	0·27	0·61	0·77
C	3·92	0·42	0·34	0·26	0·63	0·77

Thus in these tests the engine developed about 77 per cent. of the total power theoretically possible, a remarkably high proportion.

To expedite calculations of efficiency fig. 328 is given, showing the air standard values for the range of compressions usual in petrol engines. The lower dotted curve gives values equal to 0·8 of the air standard, in accordance with the general result of the author's experiments (*v.* Vol. I, p. 274).

Mr. L. G. E. Morse's experiments on the efficiency of a four-cylinder, 3·56 ins. × 5·11 ins. Daimler petrol engine of 1906 type are described in the *Proc. Inst. A.E.*, Vol. III; the experiments on thermal efficiency were directed mainly with a view to ascertain the effect of varying strength of mixture upon the performance of the engine.

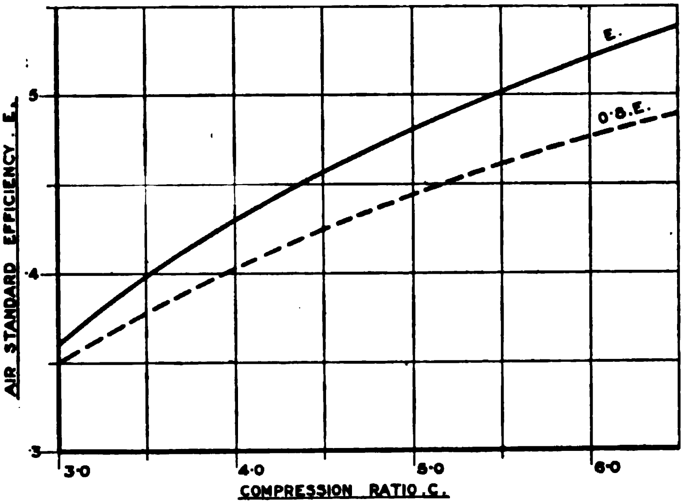

Fig. 328

From Morse's curves fig 329 has been drawn, giving full load results at 720 and at 1220 r.p.m.; it will be seen that at both speeds in these tests maximum power and maximum efficiency appear to practically occur together.

An analysis of the exhaust at the point of maximum efficiency furnished the following results:

	CO_2	CO	O	H	N
At 720 r.p.m. . .	13·5	0·7	0·2	0·0	85·6
At 1220 r.p.m. . .	14·0	0·0	0·0	0·0	86·0

From these volume percentages the ratio of air to petrol has been found by calculation (*v.* Chap. IX, on Carburettors, Eq. (9″)) to be about 13 and 14 respectively. Thus, practically speaking, these experiments showed that complete combustion, maximum power, and maximum thermal efficiency all appeared to occur simultaneously.

In the accompanying table some values of thermal efficiency for various petrol engines are collected for convenience of reference; the increase of indicated thermal efficiency with speed results from the diminished heat loss due to shorter time of contact of the working charge with the cylinder walls; the brake thermal efficiency, however, increases to a maximum and then decreases, owing to a decline in the mechanical efficiency with increase of speed.

THERMAL EFFICIENCY OF PETROL ENGINES

Engine	Bore and stroke, inches	Comp. ratio	Air stand. eff.	Revs. per min.	At — load	Thermal eff.		Rel. ind. ther. eff.	Authority
						Indic.	Brake		
1-cylinder Clèment	2·36 × 2·74	3·75	·411	1270	full	·199	·160	·485	Callendar
4-cylinder Daimler (1906)	3·56 × 5·11	—	—	720	No.	·174	0	·420	Morse
		—	—	720	half	·235	·196	·567	
		3·85	·415	720	full	·245	·227	·591	
		—	—	1220	full	·272	·223	·656	
4-cylinder Siddeley	4·62 × 5·08	4·18	·435	530	0·6	·210	·187	·483	Hopkinson
				930	full	·254	·214	·584	
4-cylinder Clément-Talbot. $\frac{\text{Air}}{\text{Petrol}} = 14$	3·35 × 4·73	3·92	·42	1250	full	·244	·198	·582	Watson
		4·35	·44	1250	full	·248	·20	·564	
		4·71	·46	1250	full	·250	·20	·543	
4-cylinder Wolseley	5·0 × 5·5	4·17	·435	1000	full	—	·216	—	Remington
4-cylinder White & Poppe	3·15 × 3·54	3·86	·421	1650	full	—	·178	—	Poppe
	3·94 × 4·33	3·78	·415	1360	full	—	·21	—	
	4·33 × 4·73	3·9	·424	1250	full	—	·22	—	
	4·73 × 5·12	4·25	·445	1150	full	—	·248	—	
	5·0 × 5·12	4·25	·445	1150	full	—	·237	—	
	3·15 × 5·12	4·7	·466	1150	full	—	·232	—	

Prof. Callendar (*Proc. Inst. A.E.*, I, p. 223, and III, pp. 259, 457, *et seq.*) has proposed an expression for the relative efficiency

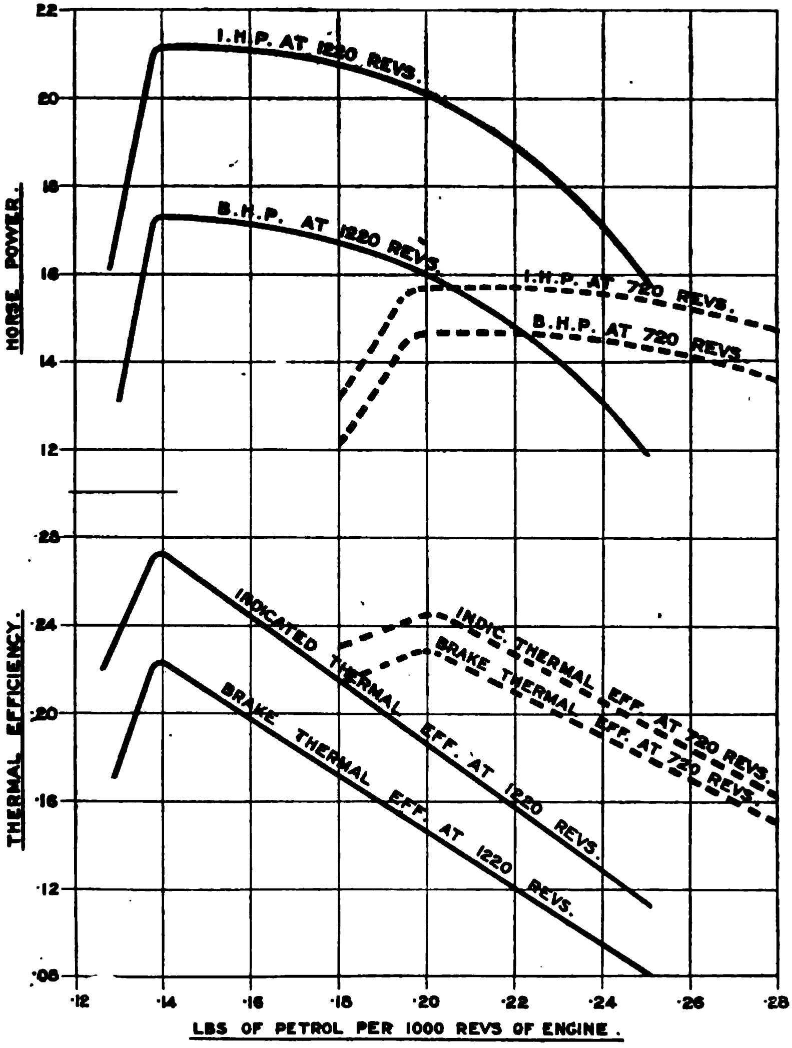

FIG. 329

of an internal combustion engine in terms of its size and speed only.

This expression is based on the reasonable assumptions that in similar engines, under similar conditions, the proportion lost of the ideally utilisable heat supplied to the engine, due to contact with the cylinder walls, varies:

(1) Directly as the surface and inversely as the volume of the combustion chamber; that is, as $\frac{d^2}{d^3}$, or $\frac{1}{d}$.

(2) Directly as the time of contact of the heated gases with the cylinder walls; that is, as $\frac{1}{n}$. And

(3) That in small high-speed engines the greater part of the loss is, to a first approximation, independent of time, and proportional to the surface volume ratio, i.e. $\frac{1}{d}$ only.

(1), (2), and (3) are symbolically expressed in the statement:

$$\text{Proportion of heat lost} = \frac{a}{d} + \frac{b}{nd}$$

where a and b are constants.

From this, using the author's result for the actual working mixture (Vol. I, p. 275), Prof. Callendar deduces finally:

$$\text{Rel. eff.} = 0{\cdot}8\left\{1 - \frac{1}{d}\left(a + \frac{b}{n}\right)\right\} \qquad (39\text{F})$$

'Corresponding speeds' for similar engines are those for which nd has the same value; as $nd = \frac{6}{r}\,\sigma$, where r is the stroke-bore ratio and σ the piston speed it is seen that similar engines are to be compared, on this theory, at the same piston speed.

Much discussion has arisen as to the proper values to be assigned to the constants a and b in Eq. (39F).

Prof. Callendar, from an examination of cases, has found the following figures:

Engine	$a =$	$b =$	Experimenter
Clèment	0·7	300	Callendar
Siddeley	0·9	370	Hopkinson
Talbot	0·8	380	Watson

It must be remembered that these engines were not really similar; this being so, the degree of agreement in the values of the constants may be regarded as satisfactory.

In his original paper Prof. Callendar pointed out that, neglecting pockets, motors might be regarded as practically similar if the stroke-bore ratio were proportioned to $(c - 1)$, where c is the volume-ratio of compression, which implies that v varies as d^3, *v.* Eq. (35); and in *Proc. Inst. A.E.*, III, p. 462, he finds that an examination of Dr. Watson's experimental results supports the assumption that the effect of pockets may be disregarded in this connection.

Formula (39F) is of much theoretical interest, and of practical importance in its bearing upon the rating of petrol engines; for an account of its application in this manner reference may be made to the original paper.

In the following table test results from 162 White & Poppe engines are given, showing the horse-power lost in the cooling water, the ratio of this to the brake horse-power, and the petrol consumption in pints per BHP hour. The piston speed throughout was about 985 f.p.m.

TEST RESULTS FROM 162 FOUR-CYLINDER WHITE & POPPE ENGINES, SHOWING BHP, COOLING LOSS, AND PETROL CONSUMPTION

Number of engines tested	Bore and stroke, inches	Valves on same or opposite sides of cylinder	Average BHP	Average HP lost in cooling	Ratio of $\frac{\text{Cooling loss}}{\text{BHP}}$	Average petrol used, pints per BHP hour
71	3·15 × 3·54	O.	19	28½	1·50	0·85
24	3·94 × 4·33	S.	33½	42½	1·27	0·72
34	4·33 × 4·73	O.	42½	57½	1·35	0·69
26	4·73 × 5·12	O.	55	62½	1·14	0·61
7	5·0 × 5·12	O.	60½	70	1·16	0·64

Power and Efficiency.—The test results just given are valuable on account of the large number of engines involved, the care with which all Mr. Poppe's tests are conducted, and the fact that the figures obtained represent the normal performance of a series of standard engines. It is of interest to deduce the values of the efficiencies, and these and other deductions are tabulated on the following page.

It will be noted that in this series of engines the value of the brake effective mean pressure increases with the size of the engine.

The figures in column 6 are calculated from the volume ratio of compression (*v.* Vol. I, pp. 244–50); the heat supply per BHP hour is obtained by multiplying the petrol consumption in pints per BHP hour by 16,800; the brake thermal efficiency is the ratio of the number 2545 to the numbers in column 7.

The figures in column 9 are the values of the ratio of corresponding figures in columns 8 and 6; no particulars as to mechanical efficiency are given; this has been accordingly assumed at a constant mean value of 0·85 throughout (*v.* Vol. I, pp. 280–2; also *ante*, p. 562).

Efficiency and Heat Expenditure of some Four-Cylinder White & Poppe Engines at a Piston Speed of about 985 Feet per Minute

1	2	3	4	5	6	7	8	9	10	11		
										Approximate per cent. of heat supply expended in:		
No of engines tested	Bore & stroke, inches	Average BHP by test	Average ηp lbs. per sq. in.	Volume ratio of compn. C.	Per cent. air stand. eff.	Heat supply in B.Th.U. per BHP hour	Per cent. brake thermal eff.	Per cent. rel. brake thermal eff.	Per cent. relative ind. eff., assuming $\eta = 0·85$	Indicated work	Heating circulating water	Exhaust gases, radiation, &c.
71	3·15 × 3·54	19	80·5	3·86	42·1	14,260	17·8	42·3	50·0	21·0	26·7	52·3
24	3·94 × 4·33	33½	90·7	3·78	41·5	12,100	21·0	50·6	59·5	24·7	26·7	48·6
34	4·33 × 4·73	42½	95·0	3·9	42·4	11,580	22·0	51·8	61·0	25·7	29·7	44·6
26	4·73 × 5·12	55	103·0	4·25	44·5	10,240	24·8	55·7	65·5	29·2	26·3	42·5
7	5·0 × 5·12	60½	101·6	4·25	44·5	10,740	23·7	53·3	62·7	27·9	27·5	44·6

1 B.Th.U. = Heat necessary to raise 1 lb. (av.) of water from 59° F. to 60° F.
Joule's equivalent, J, is taken as 778 foot-lbs. per B.Th.U.
1 Horse-power hour = $\frac{33,000 \times 60}{778}$ = 2545 B.Th.U. per hour.
Heat value of petrol at 60° F. is taken as 16,800 B.Th.U. per *pint*.

In columns 11 an approximate heat expenditure account is given; for this series, roughly speaking, one-quarter of the heat appears as indicated work, one-quarter is lost in the cooling water, and one-half in the exhaust gases, radiation, &c.

In Fig. 330 curves are exhibited showing the variation of power with speed. Mr. Poppe considers the normal power in his engines as attained at a piston speed of 5 metres per second (984 ft. per min.); the ratio of this normal to the maximum power has the following values for the five engines tested:

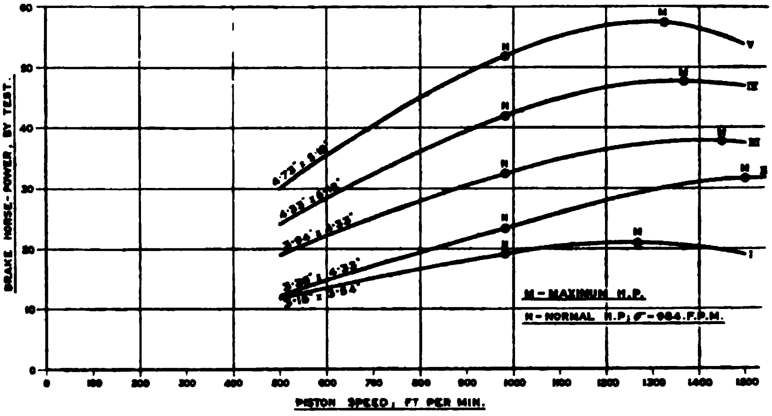

FIG. 330

Engine . .	I	II	III	IV	V
Ratio . .	0·93	0·74	0·87	0·88	0·90

No. II is anomalous, its graph being very flat compared with the remaining four; excluding this, the ratio of normal to maximum power has the average value 0·9.

The mean velocity of the gas through the inlet valve at maximum HP (*v.* Eq. (31)) is:

Engine .	I	II	III	IV	V
v = .	9400	6830	9000	9880	11,200 ft.p.m.

The low value in the case of No. II explains the anomaly just referred to; the average of the remaining four is 9870 ft. per min., which is too high a velocity (*v.* p. 534).

In the graphs of the five White & Poppe engines it will be found on trial that the maximum torque for all cases excepting No. III occurs at a lower piston speed than was reached in the tests; the graph of No. II between piston speeds of 500 and 1000 ft. per min. is practically a straight line passing through the origin, and accordingly the torque and ηp are constant and at a maximum value throughout this range.

The five curves just given are for ordinary touring car engines; in the following diagram power-speed graphs are given for some racing type engines having large valves and very light reciprocating parts

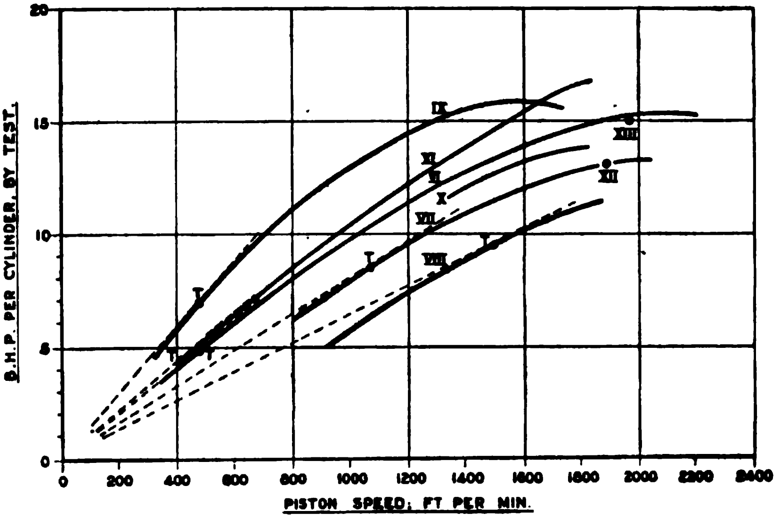

FIG. 331

(fig. 331). The point of maximum torque in each case is marked 'T'; the following table gives some further particulars of these engines:

RACING TYPE PETROL ENGINES

Engine	Date of construction	Bore d, inches	Stroke s, inches	Valve δ, inches	Lift λ, inches	Max. BHP per cyl.	Piston speed at max. BHP, ft. per min.	v at max. BHP, ft. per min.
VIII	1909	3·15	5·12	—	—	11·7	1875	—
XII	1910	3·54	4·73	—	—	13·0	1890	—
XIII	1910	3·54	4·73	—	—	15·0	1970	—
VII	1909	3·74	5·32	1·97	0·32	13·5	2020	7300
X	1908	4·0	7·0	2·25	0·63	14·2	1820	5750
VI	1909	4·0	5·5	1·88	0·31	15·4	2200	9950
XI	1908	4·06	6·0	—	—	16·5	1800	—
IX	1908	4·92	5·0	—	—	16·0	1550	—

Engine No. XIII is an improved No XII, mainly by reduction in the mass of the reciprocating parts.

CHAPTER VIII

SOME PETROL ENGINES DESCRIBED

The Four-Cylinder, 40-*HP Wolseley Engine.*—Longitudinal and transverse sections are shown in fig. 332.

The bore is 5 ins.; stroke 5½ ins.; ratio of stroke to bore 1·1.

Rating for taxation purposes = 40.

The cylinders are cast in pairs, the distance from centre to centre of the members of each pair being 5½ ins. only; the cylinder walls are $\frac{5}{16}$ in. thick, finished size; thus the cylinders of each pair are placed as close together as possible.

The combustion chambers are flat-topped; the valves are side by side in one pocket on the carburettor side of the engine; this pocket is $\frac{7}{8}$ in. in depth; the form of the combustion chamber is well shown in the transverse section.

The valve stems, tappet rods, and tappet rod guides are neatly enclosed by light, easily removable metal covers. The clearance between tappets and valve stems is adjustable; pressure contact between the lower ends of the tappets and the cam rollers is maintained by helical springs housed in the tappet guides as shown.

The volume ratio of compression c is 4·17; the corresponding compression pressure from Eq. (37) is 98½ lbs. per sq. in. absolute; tested by an Okill compressometer at about 1000 revolutions per min., a compression pressure of about 100 lbs. per sq. in. absolute was recorded.

The volume of the combustion chamber (*v.* Eq. 35) is 34 cub. ins., and the total surface is 98·5 sq. ins., of which 69·0 sq. ins. is water-cooled and 29·5 sq. ins. (including the piston and valve crowns and sparking-plug) is uncooled area.

The ratio of surface to volume is therefore $\frac{98\cdot5}{34} = 2\cdot9$; a hemispherical chamber having the same cubic content would have a ratio of surface to volume of 1·78 only. The total surface exposed to the working gases when the piston is at the bottom of its stroke is 185 sq. ins.; hence the ratio of combustion chamber

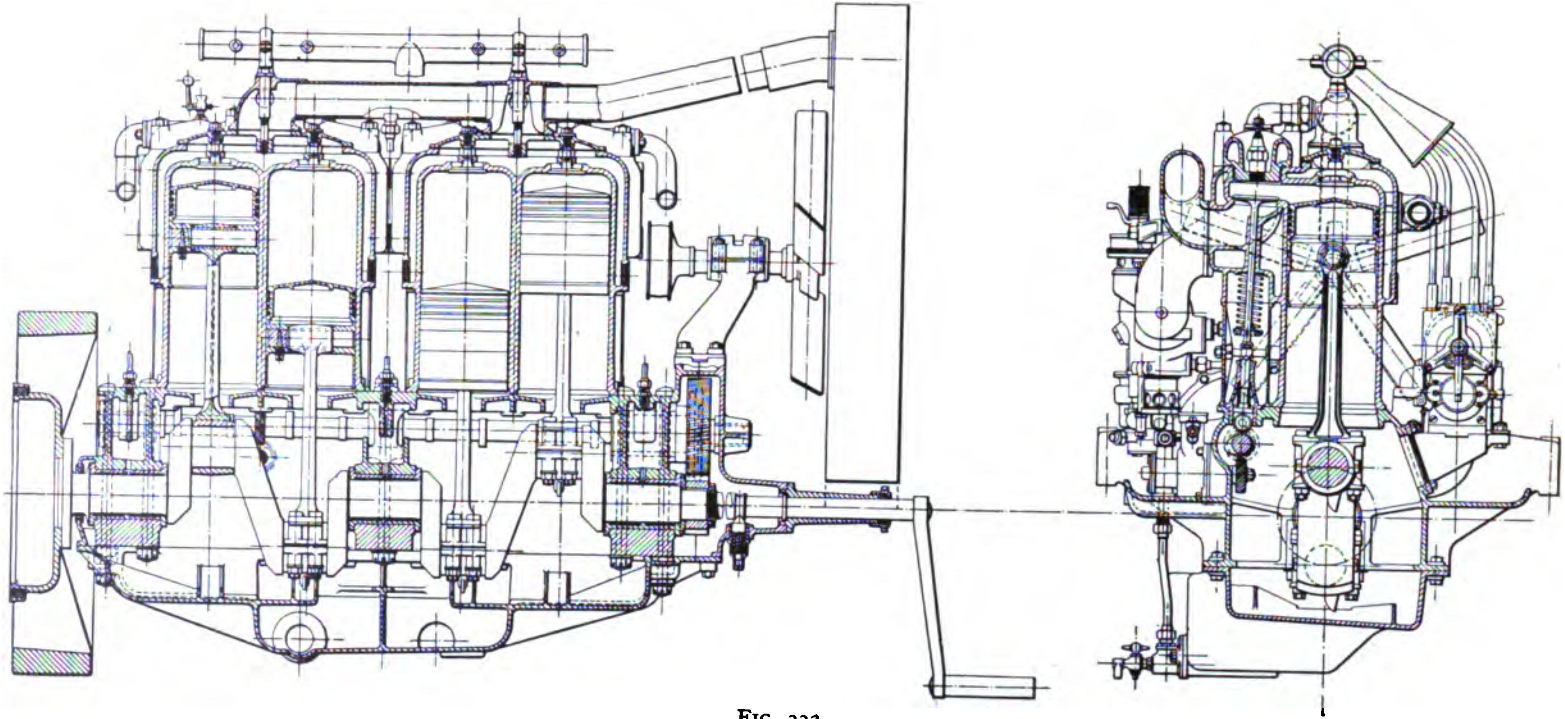

FIG. 332

surface to total surface exposed is $\frac{98\cdot5}{185} = 0\cdot53$. The ratio of uncooled to total surface in the combustion chamber is $\frac{29\cdot5}{98\cdot5} = 0\cdot3$.

The wall of the pocket approaches to within ¼ in. of the edge of the exhaust valve; some experiments that have come under the authors' notice tend to the conclusion that in such cases a considerable part of the valve periphery—sometimes approaching one-half even—may be comparatively ineffective as a means of discharge for the burnt gases.

The jackets extend about half-way down the cylinders, and the exhaust valve seatings are well cooled. The circulation is maintained by a centrifugal water pump direct driven by the engine; the radiator is of the Zimmerman type, with tubes about ¼ in. external diameter; it is 4 ins. deep, and its frontal area is 507 sq. ins., with a belt-driven five-vane fan behind, as shown in the longitudinal view.

In the transverse section the water-cooled valve covers are clearly indicated; these prevent overheating of the combustion chamber surfaces immediately above the valves, and thus reduce the risk of pre-ignition.

The poppet inlet and exhaust valves have each a diameter of 2·375 ins. in the throat; the lift of the inlets is 0·28 in., and of the exhausts 0·38 in.

Thus the ratio of $\frac{\delta}{d}$ is 0·475; and of $\frac{\lambda}{\delta}$ 0·118 for the inlet valves and 0·160 for the exhausts.

The piston speed at maximum power is about 1550 ft. per min. by test; hence by Eq. (31) the mean velocity of the gas through the inlets at maximum power is $v = 6850$ ft. per min., a rather low value, suggesting that part only of the valve area is effective, for the reason already stated.

The normal speed of the engine is considered to be 1000 r.p.m., corresponding to a piston speed of 917 ft. per min., and a mean velocity of gas through the inlet valves of only about 4050 ft. per min. By test the BHP at this speed is 49.

The valve timing is as follows:

Inlets: open 11° late close 19° late.
Exhausts: open 38° early close 7° late.

The pistons are of 40-ton pressed steel with slightly coned tops and an annular stiffening and cooling rib on the under side; four spring rings are fitted in each. They are 5½ ins. in length, machined all over, and weigh, complete with rings and gudgeon, 5·94 lbs. each. The gudgeons are of steel, ⅞ in. diameter, fixed in the piston bosses by a single, locked, steel set screw with long coned point. The length of the gudgeon bearing is 2½ ins.

At the maximum explosion pressure of about 300–320 lbs. per sq. in. the gudgeon sustains a momentary load of roundly 6000 lbs., corresponding to 2750 lbs. per sq. in. of projected area of gudgeon bearing.

The connecting-rods are tee-ended steel stampings of the usual **H**-section, $11\frac{1}{2}$ ins. long from centre to centre; the ratio of this to the length of crank is $\frac{11\frac{1}{2}}{2\frac{3}{4}} = 4{\cdot}18$. The rods swing in slots in oil baffles formed in the tops of the crank-chambers.

The weight of one rod complete is 6·8lbs., made up, when weighed in a horizontal position, of 1·6 lbs. at the small end and 5·2 lbs. at the big end. The gudgeon, wrist pin, or small-end bearing is $\frac{7}{8}$ in. diameter × $2\frac{1}{2}$ ins. long, bronze on steel; the crank-pin or big-end bearings are of white metal, $2\frac{1}{4}$ ins. internal diameter × $2\frac{1}{2}$ ins. long. The big ends are built up on the tee end of the connecting-rod, with four $\frac{9}{16}$ in. bolts in each.

The maximum pressure per sq. in. of projected bearing area is $\frac{6000}{2\frac{1}{4} \times 2\frac{1}{2}} = 1070$ lbs. per sq. in. in the big ends, which is low, the figure in many case being roundly 1500 lbs. per sq. in.; this is due partly to the large crank-pin diameter necessary for crankshaft stiffness; the centre line of the connecting-rod does not bisect the crank-pin bearing, but divides it in the ratio of $1\frac{1}{8}$ ins. to $1\frac{3}{8}$ ins.; this is a practice not infrequently met with in small petrol engine design, though regarded with disfavour by steam engineers; it causes an unequal distribution of the bearing pressure on the crank-pin, as indicated in the following diagram (fig. 333), and this inequality is augmented by any flexure which may occur in the crankshaft under the load due to the explosion pressure.

Referring to fig. 333, if the connecting-rod axis divide the big-end bearing into the portions *a* and *b*, then it may easily be shown that, D being the crank-pin diameter in inches, and P the total load on the bearing in lbs.:

$$\left.\begin{aligned}\text{Bearing pressure at A} &= \frac{2b - a}{(a + b)^2} \cdot \frac{2\text{P}}{\text{D}} \text{ lbs. per sq. in.}\\ \text{Bearing pressure at B} &= \frac{2a - b}{(a + b)^2} \cdot \frac{2\text{P}}{\text{D}} \text{ lbs. per sq. in.}\end{aligned}\right\} \quad (40)$$

thus the ratio $\frac{\text{BD}}{\text{AC}} = \frac{2a - b}{2b - a}$.

In this case the pressures at A and B are roundly 750 and 1400 lbs. per sq. in. respectively, and their ratio is as 13 to 7.

On this account in cases where the axis of the connecting-rod does not bisect the crank-pin length, it appears better practice to make the big end of the connecting-rod symmetrical, and leave a collar, C, on the crank-pin, as indicated in fig. 334; this is a mode of design frequently adopted.

The crankshaft is of 'Vickers' crankshaft and axle steel,' and is carried in three long bearings; it is of substantial proportions, being 2⅛ ins. in diameter; the crank-pins are 2¼ ins. in diameter.

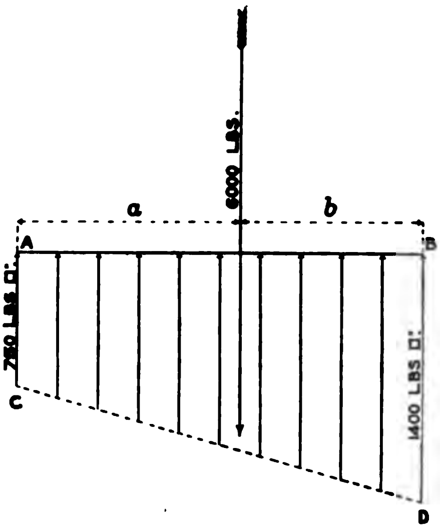

FIG. 333

The bearings are lined with white metal; the forward bearing is 4½ ins. in length, the central bearing 4 ins., and the after bearing (flywheel end) is 4⅜ ins.; the bottom cover of the crank-chamber can be removed without disturbing these main bearings.

The after end of the crankshaft has a flange and centering spigot, and to this flange the flywheel and clutch case are bolted.

The flywheel is 20½ ins. in diameter, with a face width of 4½ ins.; the weight of the rim is 91 lbs.; at 1000 r.p.m. the energy of the rim is about 10,000 foot-lbs.

The half-speed or timing shaft is driven by helical toothed gearing from the forward end of the crankshaft.

The oil pumps, on the carburettor side, are actuated by skew gearing from the half-speed shaft; the centrifugal water pump and magneto are similarly driven, and placed on the other side of the engine.

The ignition is by a hand-controlled high-tension Bosch dual magneto (*v.* Chap. III.); the sparking-plug is fitted in the water-cooled cover over the inlet valve; when a second plug is required it is placed above the exhaust valve, as indicated in the transverse section. The order of firing is 1342.

The carburettor is of the Wolseley Company's own design, and has three jets. The first, which is very minute, discharges into a small mixture pipe, and is brought into action by the throttle control gear only when the engine is running very slowly, as e.g. in dense town traffic. A further throttle opening cuts out this first jet and opens to a second, so proportioned as to give economical running under normal conditions of work. When the throttle is at its full opening a third jet comes into play, in conjunction with the second; this third jet is large in proportion to the mixture pipe, and thus increases the proportion of petrol to air. At full throttle, power is regarded as the primary requisite, high economy becoming a secondary consideration.

Hand control of the auxiliary air valve spring is provided, so that the driver can vary the richness of the mixture supplied to the engine at will.

Petrol is supplied to the carburettor under pressure maintained by a small air pump driven by the engine; the petrol tank has a capacity of 22 gallons.

The lubrication is semi-mechanical. Oil contained in a reservoir at the side of the engine is supplied by a small pump drawing from the engine sump; a second pump delivers the oil from this reservoir to the main shaft bearings and big-end troughs; the oil supply pipes to the bearings and the troughs are clearly shown in the longitudinal section (fig. 332). The big ends are lubricated from the narrow troughs by aid of the oil scoops attached to their bottom brasses as shown; the remaining engine bearings depend upon the splash. A baffle is cast in the crank-chamber above each crank to intercept the oil whirled from the crank cheeks, and prevent over-lubrication of the cylinder surfaces.

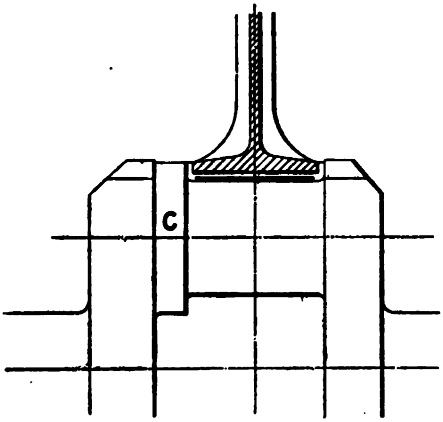

FIG. 334

The weight of the engine complete with radiator, fan, pumps, carburettor, magneto, and flywheel, in running order, but without fuel or water, is 850 lbs. Without the flywheel the weight is 728 lbs. This gives the following figures per horse-power:

	At BHP	
	Normal (49)	Max. (60)
Weight in lbs. per BHP complete	17·4	14·2
„ „ „ without flywheel	14·9	12·1

In fig. 335 the graph of the power and speed for this engine is given; the maximum BHP is roundly 60, attained at about 1700 r.p.m.; the maximum speed for short periods of running is about 1800 r.p.m.;

the normal speed is considered to be 1000 r.p.m., and at this speed the BHP is 49; hence the ratio of normal to maximum power is $\frac{49}{60} = 0{\cdot}82$.

The petrol consumption averages 0·7 pint (about 0·63 lb.) per BHP hour at normal speed; the petrol used in tests was Pratt's, specific gravity 0·72; heat value 16,800 B.Th.U. (° F.) per pint at 60° F.; hence the heat supplied per BHP hour at normal speed is 11,760 B.Th.U. The brake thermal efficiency is consequently $\frac{2545}{11{,}760} = 0{\cdot}216$.

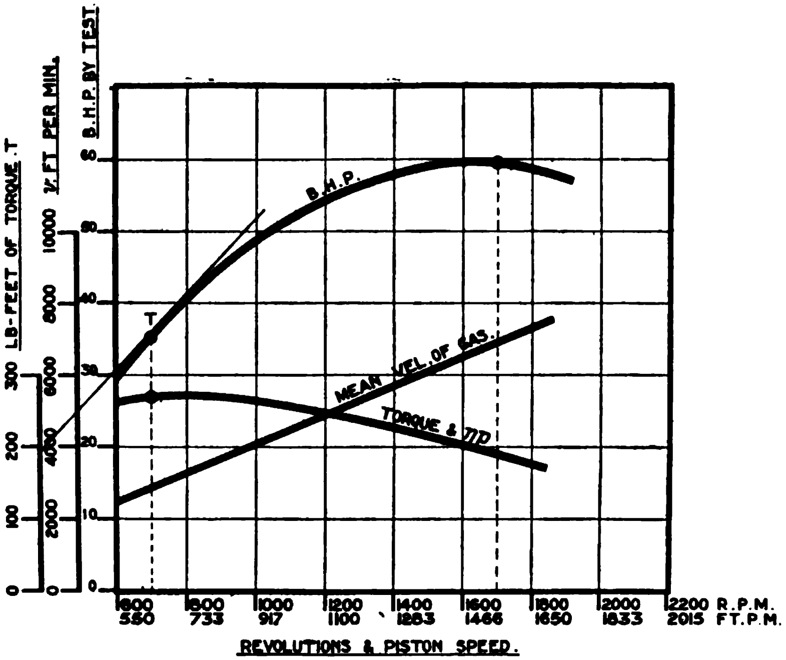

FIG. 335

The volume ratio of compression is 4·17; hence (Vol. I, p. 248) the corresponding air standard efficiency is 0·435; the brake relative efficiency is therefore 0·497.

Allowing that the maximum possible efficiency is 0·8 of the air standard (*v.* Vol. I, p. 274), the absolute brake thermal efficiency of this engine is $\frac{0{\cdot}216}{0{\cdot}435 \times 0{\cdot}8} = 0{\cdot}622$.

On fig. 335 is also shown the variation of torque and ηp with speed (*v.* Eqs. (38) and (39), *ante*); the maximum torque occurs at the low speed of 700 r.p.m. The line of mean velocity of gas through inlet valves for assumed full charge is also given.

BHP by test	Revs. per min.	σ ft. p. m.	T lb.-feet, Eq. (38)	ηp lbs. per sq. in., Eq. (39)
30	600	500	262	91·5
36	700	642	270	94·3
49	1000	917	257	89·8
60	1700	1560	185	64·7

Aeroplane Motors.—Within the past few years a large number of exceedingly light petrol-driven engines have appeared, designed for the purpose of propelling air craft. In the *Automotor Journal* for October 8, 1910, Mr. J. S. Critchley has tabulated particulars of no fewer than thirty-five different designs, including seventy-six engines ranging in size from the vertical four-cylinder, 118 HP, water-cooled Clèment, having a bore of 7·5 ins. and stroke of 9 ins., and weighing about 9¼ lbs. per HP, down to the eight-cylinder, diagonal, water-cooled Antoinette engine of 3·15 ins. bore and 3·15 ins. stroke, with a power rating (by S.M.M.T. formula (22)) of 32, and stated to weigh but 2·9 lbs. per HP at this figure.

The engines are classified as of vertical, diagonal, horizontal, radial, and rotary types; about two-thirds of the makers employ water cooling, the remaining one-third being air cooled.

The number of cylinders varies from two to sixteen, the latter number being found in two of the diagonal-type Antoinette engines. Cases of engines with two, three, four, five, six, seven, eight, ten, fourteen, and sixteen cylinders are given.

Engine weights per rated horse-power are stated to vary from 20·7 lbs. down to 1·8 lbs., the latter being the figure given for the fourteen-cylinder, 123 HP, rotary Gnome engine.

Mr. F. W. Lanchester has given the useful figures in the table on p. 582, reproduced in the *Autocar* of September 3, 1910, in connection with the Antoinette, Renault, and Gnome aero engines, and—for purposes of comparison—corresponding results for the 22 HP and 38 HP standard Daimler-Knight car engines.

The weight is defined to be the weight of the engine in running order, including all accessories, with radiator and contained water (where fitted), but excluding silencer, flywheel, and petrol and oil tanks.

The low mean pressure and large petrol consumption of these aero engines will be noted; well-designed car engines of from 30 to 60 HP require from about 0·5 to 0·7 *pound* of petrol per BHP hour, a good average value being 0·6. The average of the three engines in

Mr. Lanchester's table is 0·91, that is, 50 per cent. in excess of normal car engine consumption. The primary consideration, however, has so far been reduction in weight in this class of engine.

Item	Antoinette	Renault	Gnome	Daimler	
				1	2
Nominal HP . . .	50	55	50	22	38
Actual BHP . . .	54·3	58·8	40	39	57·25
Av. revolutions per min. .	1200	1800	1200	1400	1200
Bore (inches) . . .	4·35	3·44	4·35	3·8	4·9
Stroke (inches) . .	4·15	4·75	4·73	5·14	5·14
Number of cylinders .	8	8	7	4	4
Mean pressure as shown by brake	72·5	73·6	58·3	94·5	97
Weight as defined (lbs.) .	292·4	375	165	360	520
Weight per BHP (lbs.) .	5·38	6·4	4·1	9·2	9·1
Extra for radiator, not included, allow . .	—	—	—	1·0	1·0
Petrol per BHP (per hour, lbs.) . . .	0·94	0·96	0·89	0·67	0·54

The 50 *HP Gnome Rotary Engine.*—An interesting example of the rotary, air-cooled aeroplane engine is furnished by the seven-cylinder, 50 HP Gnome design of MM. Seguin, of which outside views are given in the accompanying fig. 336.

In this engine the seven cylinders are arranged at equal angular intervals, and, together with the crank-chamber, rotate around a fixed crankshaft. The seven connecting-rods, all in the same plane, operate on one crank-pin, as shown in fig. 337; one of the rods is rigidly connected to two large rings of L-section having six pairs of holes drilled in them at angular distances apart of 51$\frac{3}{7}$°. These pairs of holes severally receive the six pins connecting up the big ends of the remaining six connecting-rods to the crank-pin. One rod is necessarily rigidly connected to this large double-ring cage in order to define its position. The cage carries two ball races through which the actions of the several rods are communicated to the fixed crank-pin. With this arrangement the angularity of the six shorter rods slightly exceeds that of the seventh rigid rod; but this difference does not appear to affect the practical running of the engine. The rods are of steel, of H-section as shown, with milled surfaces.

In order to obtain the greatest economy of material the extreme course is adopted of machining the cylinders from solid steel ingots; the finished thickness of the cylinder walls is only 0·059 in. (1½ mm.), and the necessary strength is supplied by the cooling fins formed on the outer surface, which increase in diameter towards the combustion

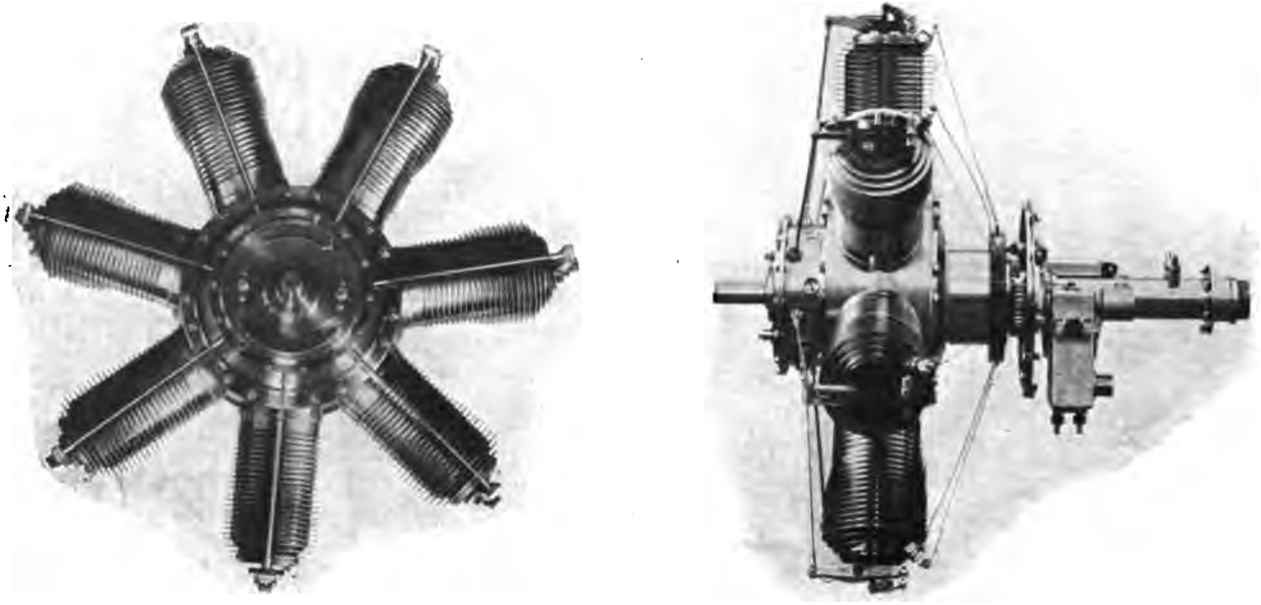

Fig. 336

chamber end as shown in the sectional view in fig. 337. The removable cylinder tops contain the exhaust valve seats, which are screwed into place by aid of a special spanner.

The finished cylinders are 4·35 ins. in bore ; the stroke is 4·73 ins.

An ingenious method of attaching the cylinders to the crank-chamber is adopted :

Each cylinder is turned perfectly cylindrical outside, and bears at its lower end a spring ring groove similar to that of a piston ; the seven cylinders are then pressed into circular holes bored in the crank-chamber. When in position a contracting spring ring is sprung tightly into each groove ; these rings when in place stand out beyond the outer surface of the cylinders and take an end bearing on turned faces provided for the purpose within the crank-chamber. To prevent any motion of the spring rings, seven long stout pins are passed through the crank-chamber as shown at C C C in fig. 338, each pin bearing against the rings of two adjacent cylinders ; the cylinders are themselves prevented from revolving in their seatings by a small ' snug ' between their bottom ends and the crank-chamber. The details of the cylinder attachment are clearly shown in the two views, figs. 337 and 338.

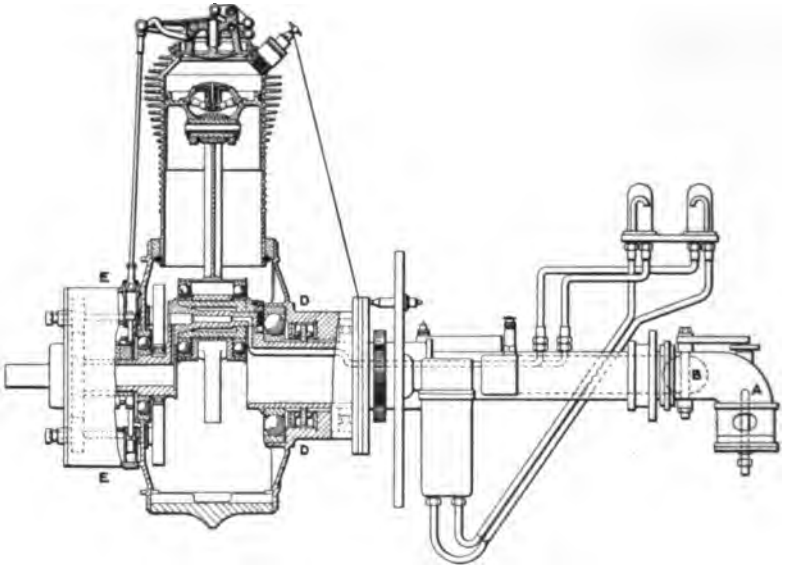

FIG. 337

The crank-chamber is cylindrical in form and built up of nickel steel ; no aluminium is used. It is borne on the fixed crankshaft by a pair of ball bearings housed in the end plates ; the driving end plate D carries a cylindrical extension 5⅛ ins. in diameter, to which the propeller boss is attached by means of two keys.

Through the hollow fixed crankshaft the working charge is introduced into the crank-chamber ; the hollow shaft also permits the two copper lubricating pipes to be led into the centre of the engine (fig. 337).

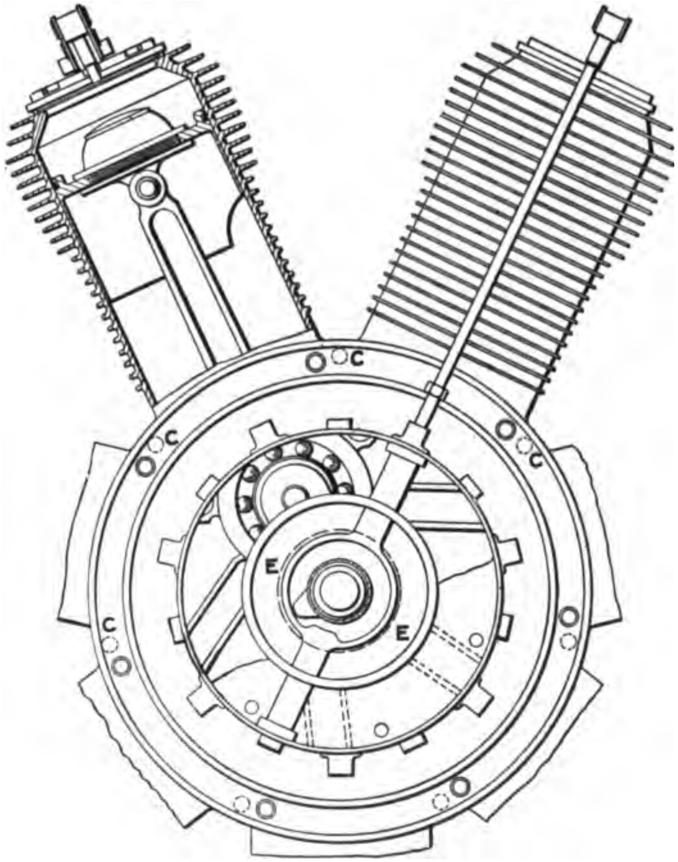

FIG. 338

The inlet valves are of the automatic type formerly general in petrol engines, but now entirely superseded by the cam-lifted type. The automatic valve is here used partly to save weight, and partly because its adoption permits the valve to be placed in the centre of the piston, as shown in fig. 337. They are made as light as possible, the stems even being hollow for part of their length, and are counter-weighted to balance the centrifugal force due to the rapid rate of revolution.

Ordinarily the centre of the piston is one of the hottest parts of an engine, but the valves are here kept from becoming overheated by the constant passage through them of the fresh cool mixture from the crank-chamber into the combustion space.

The exhaust valves are placed in the cylinder heads and are also balanced against centrifugal actions; they are mechanically operated by light tension rods actuated by seven internal cam rings driven by a single solid cam of normal type rotating at half engine speed relatively to them; the whole cam device is contained in the cylindrical casing E E (figs. 337 and 338); the exhaust gases are discharged directly into the atmosphere.

The very light steel pistons are fitted sometimes with one ring only, as shown in fig. 337, and sometimes with three; these rings are compound, each consisting of an L-shaped thin gun-metal ring with a stiff steel Ramsbottom spring ring behind it. The gun-metal rings are very thin, and are considered to act similarly to a cup-leather in preserving gas-tightness.

Mr. R. W. A. Brewer, whose description of the Gnome engine [1] has been of assistance in this connection, points out that compound rings of this type are apt to accumulate carbon and then become stuck, with resulting leakage and loss of compression, and he states that after some hours working the compression is sometimes found to have fallen considerably below the normal.

The single-spray carburettor A, shown on the extreme right of fig. 337, has no float chamber, the petrol supply being regulated by the driver by means of a screw-down valve; the engine speed is regulated by a disc throttle valve, B, placed at the junction of the carburettor and crankshaft.

Ignition is by high-tension magneto; the armature gives two firing currents per revolution, while the engine requires seven in two revolutions, or $3\frac{1}{2}$ per revolution; the armature is accordingly so geared as to run at $\frac{3\frac{1}{2}}{2} = 1\frac{3}{4}$ times the revolution rate of the engine cylinders.

If the working impulses of the several cylinders are to occur at equal angular intervals throughout each revolution, then there must be an odd number of cylinders in engines of this type, as Dr. Watson has pointed out. For as the cycle of each cylinder occupies two revolutions, if the cylinders were set to fire consecutively, all would fire during one revolution and none during the following. Hence the cylinders cannot fire consecutively. If they are set to fire alternately, then in order to get completely round by equal angular intervals in two turns, there must be an odd number of cylinders; this will appear clear from an examination of the accompanying fig. 339 for the case of seven cylinders, which are numbered in order of firing.

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[1] Vide *The Automobile Engineer*, July 1910.

Lubrication has, so far, presented great difficulties in the case of rotary petrol engines, and in order to obtain satisfactory running a large amount of oil must be used. In the Gnome engine castor oil is employed as the lubricant; in one test the oil consumption per hour amounted to 1½ gallons. The surplus oil is whirled from the engine and discharged with the exhaust gases directly into the atmosphere from the cylinder heads.

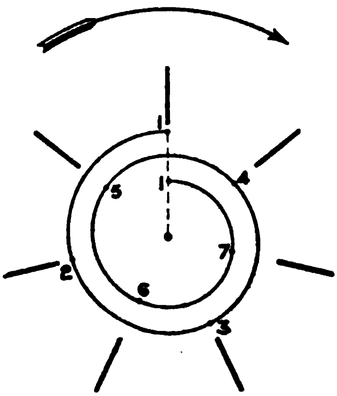

FIG. 339

The normal full speed of the engine is considered to be 1000 r.p.m., though it may be run up to 1200 r.p.m. As fitted to a Blèriot monoplane, the engine is said to run comfortably, and develop sufficient power for ordinary service at about 600 r.p.m.; by throttling down it can be run at as low a speed as 200 f.p.m. By test the 50 HP, seven-cylinder engine gives about 46 effective HP at 1000 r.p.m.; at this speed about 5 HP is expended in overcoming the air resistance to the motion of the cylinders. Mr. Brewer states that the petrol consumption is then only about 0·7 pint per effective horse-power hour, which is an economical performance. On the other hand, the oil consumption amounts to over ¼ pint per effective horse-power hour, i.e. about nine times as much as in a good car engine of the same power.

The weight of the complete engine is but 167 lbs., or at the rate of $\frac{167}{46} = 3·6$ lbs. per effective HP, which is a rather lower figure than that given in Mr. Lanchester's table (*ante*, p. 582); in that table, however, the actual BHP is taken roundly at 40, and the weight as 165 lbs., the corresponding figure for the weight per effective HP being then 4·1 lbs.

Great ingenuity is evidenced in the design and construction of the Gnome engine, but apart from its lightness it is not clear that any theoretic advantage is gained by the adoption of the rotary type. Considerable power is expended in overcoming the air resistance to the motion of the cylinders, and the difficulties of lubrication have already been pointed out.

The maximum stated piston speed, viz. at 1200 r.p.m., is only 950 f.p.m. or not more than the normal in an ordinary touring car engine;

this falls very far short of 2000 f.p.m. as attained by many car engines, e.g. the 16–20 HP 'Sunbeam,' 20 HP Crossley, 15·9 White & Poppe, 20·5 HP Vauxhall, &c.

The power output per cylinder at 1000 r.p.m. is only $\frac{46}{7} = 6{\cdot}6$.

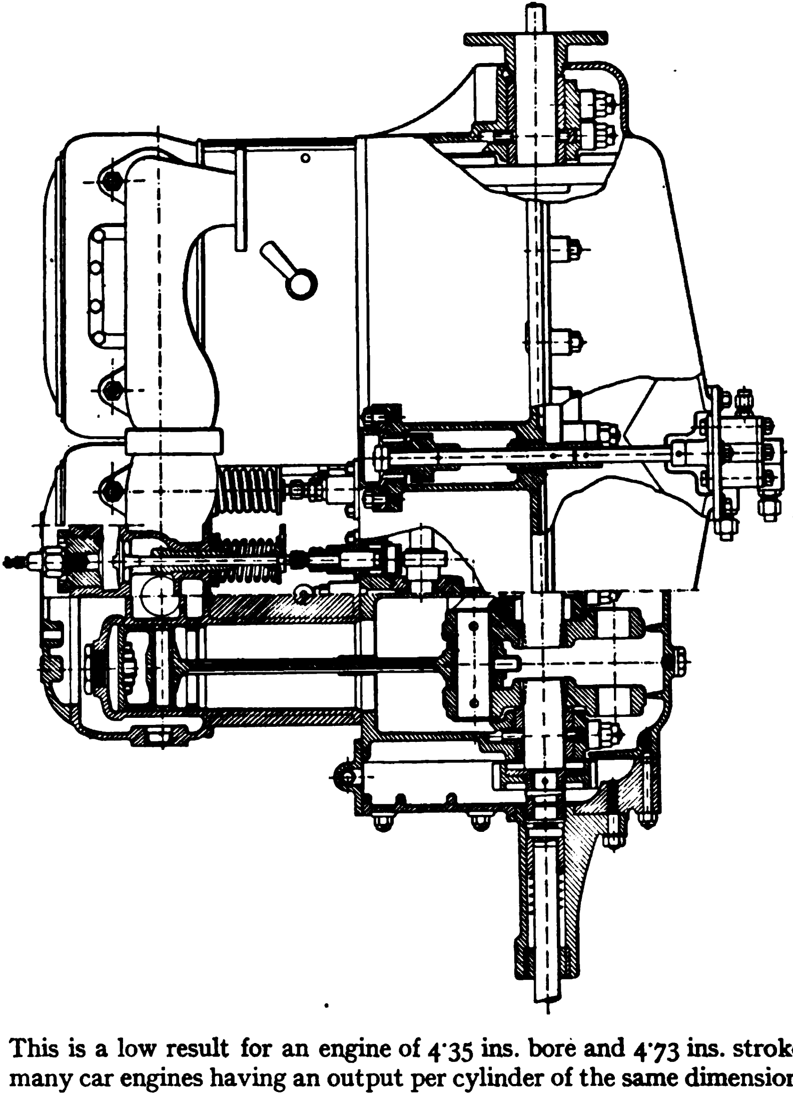

FIG. 340A

This is a low result for an engine of 4·35 ins. bore and 4·73 ins. stroke, many car engines having an output per cylinder of the same dimensions nearly twice as great.

The brake mean effective pressure corresponding to 46 effective

HP at 1000 r.p.m. from seven cylinders is 74·3 lbs. per sq. in.; this again is a low result compared with many car engine performances.

It is possible that the cylinders of the Gnome engine are kept too cool, and that better power results would be obtained by keeping the cylinders hotter. In some experiments made by the E. R. Thomas Motor Co. on a six-cylinder, 4½ ins. × 5½ ins. engine it was found that with a jacket water temperature of 100° F. the BHP was 51, and that this was increased to 62 by keeping the jacket water at 200° F. (*v. The Motor*, January 24, 1911).

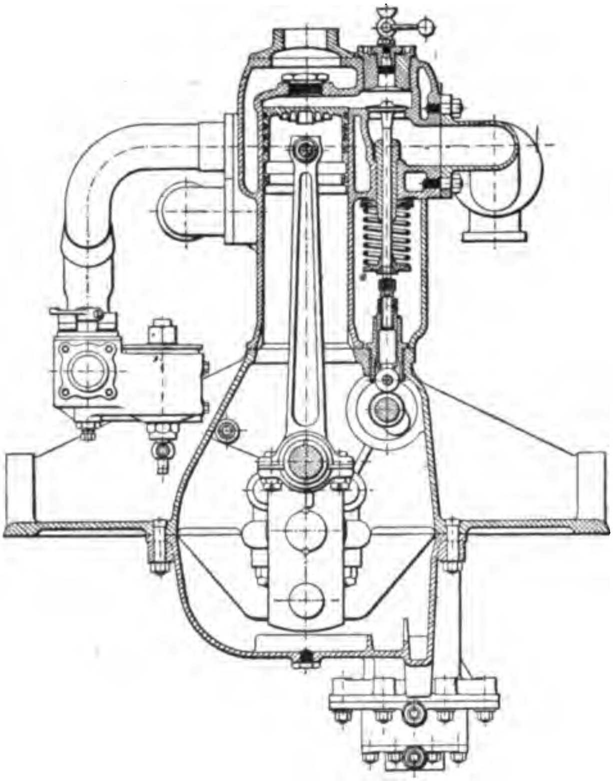

FIG. 340B

Much as the weight per BHP has been reduced in these rotary engines, it yet seems probable that figures as good may be obtained with the normal fixed cylinder type when designed and constructed with the same skill and care as are evidenced in the Gnome engine.

The 15·9 *HP White & Poppe Engine.*—Longitudinal and transverse sections are shown in figs. 340 A and B.

The bore is 3·15 ins., stroke 5·12 ins.; the ratio of stroke to bore is 1·625. Rating for taxation purposes 15·9.

The cylinders are in pairs, the distance from centre to centre of the members in each pair being $5\frac{1}{4}$ ins.; thus there is an ample water space between the cylinders, which are not placed as close together as possible on account of the crankshaft being borne in five bearings in all Messrs. White & Poppe's four-cylinder designs.

The cylinder walls are $\frac{3}{16}$ in. in thickness, the combustion chamber head being $\frac{1}{4}$ in. thick, and slightly domed; the valves are in a pocket $\frac{5}{8}$ in. in depth on one side of the cylinders.

The valve stems, tappet rods, and tappet guides are neatly enclosed by light, easily removable, metal covers.

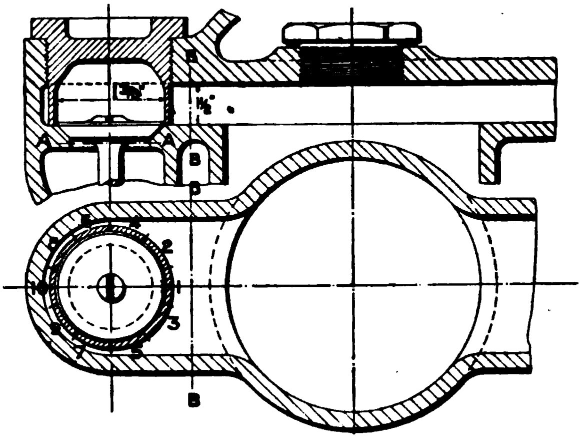

FIG. 341

The volume ratio of compression, c, is 4·7; when steel pistons are used the value is raised to 5·34; the compression pressure from Eq. (37) is 115·7 lbs. per sq. in. absolute for the normal ratio of 4·7.

The volume of the combustion chamber is 10·7 cub. ins., and its total surface is 50 sq. ins., of which 34 sq. ins. are water cooled and 16 are not.

The ratio of surface to volume is therefore $\frac{50}{10\cdot7} = 4\cdot7$; a hemispherical chamber of the same cubical content would have a ratio of surface to volume of 2·62 only.

The total surface exposed to the working gases when the piston is at the bottom of its stroke is 101 sq. ins.; hence the ratio of combustion chamber surface to total surface exposed is $\frac{50}{101} = 0\cdot5$ (app.).

The ratio of uncooled to total combustion chamber surface is $\frac{16}{50} = 0·32$.

The wall of the pocket approaches to within about $\frac{8}{16}$ in. of the edge of the valve. In this engine both valves are contained in one pocket, though in many of the White & Poppe designs the valves are on opposite sides of the engine, this disposition giving, in the opinion of the makers, a cleaner and better scavenged engine.

Some experiments by Mr. Poppe in this connection may here be conveniently referred to:—

He surrounded the inlet valves of a four-cylinder, 80 × 90 mm. (3·15 ins. × 3·54 ins.) engine with a thin cylindrical tube as shown in the accompanying fig. 341 in elevation and plan. Two series of tests were then made with a carefully calibrated air fan dynamometer (HP varying then as n^3) with one-tenth, two-tenths, &c., of the periphery of this thin tube successively removed, as indicated in the plan view.

The curves in the following diagram (fig. 342) show the results obtained; in the upper curve the test was carried to a piston speed of 980 f.p.m., at which 18 BHP was developed; in the lower the piston speed was raised to 1162 f.p.m., the BHP then (due to wire-drawing mainly) being only 15·2. It will be noted that in each series after six-tenths of the periphery of the tube had been removed a small increase only in speed and power was obtained, suggesting that the effectiveness of the valve opening adjacent to the pocket wall was small.

MR. POPPE'S TESTS.

No. of tenths removed	SERIES 1				SERIES 2			
	$\sigma =$ ft. per. min.	BHP by test	ηp lbs. per sq. in.	$v =$ ft. per. min.	$\sigma =$ ft. per. min.	BHP by test	ηp lbs. per sq. in.	$v =$ ft. per. min.
1	660	5·7	36·6	23,800	803	3·8	20·1	29,000
2	803	9·9	52·3	14,500	980	8·9	38·6	17,700
3	885	13·4	64·2	10,650	1062	11·4	45·5	12,800
4	933	15·5	70·5	8980	1086	12·6	49·2	10,460
5	950	16·5	73·5	9150	1116	13·4	50·9	10,750
6	956	16·8	74·5	9200	1132	14·0	52·4	10,900
7	962	17·2	75·8	9250	1145	14·4	53·3	11,030
8	968	17·5	76·8	9325	1150	14·7	54·2	11,080
9	974	17·8	77·5	9380	1156	15·0	55·0	11,150
10	980	18·0	78·0	9440	1162	15·2	55·5	11,200

The inlet valves were $1\frac{1}{16}$ in. in diameter in the throat, the valve stems being about $\frac{9}{32}$ in. in dameter; the lift was about 0·27 in.

Hence the nett area of the throat (A A in fig. 341) was 0·82 sq. in.; of the cylindrical lift surface, $\pi\delta\lambda$, 0·9 sq. in.; and of the cross-sectional area of the pocket, B B, 0·81 sq. in.

The thin cylindrical tube surrounding the inlet valve was $1\frac{3}{8}$ ins. internal diameter and $\frac{1}{2}$ in. long; the total surface was therefore $\pi \times 1\frac{3}{8} \times \frac{1}{2} = 2{\cdot}16$ sq. ins.; thus each one-tenth corresponded to an addition of 0·216 sq. in. to the area. In the accompanying table corresponding values of the piston speed, σ, the BHP, and the values of ηp, are given for the two series of tests; see also fig. 343. The piston speed is 12σ ins. per minute; one-fourth of this, or 3σ, should be filled with fresh gas in one-quarter of a minute; thus $3\sigma \times \frac{\pi}{4} d^2$ cub. ins. of fresh gas should enter the cylinder per quarter minute; if this gas pass through an area A sq. ins., the mean velocity is $\frac{\frac{3}{4}\sigma\pi d^2}{A}$ ins. per quarter minute, or $\frac{4}{12}$ of this in

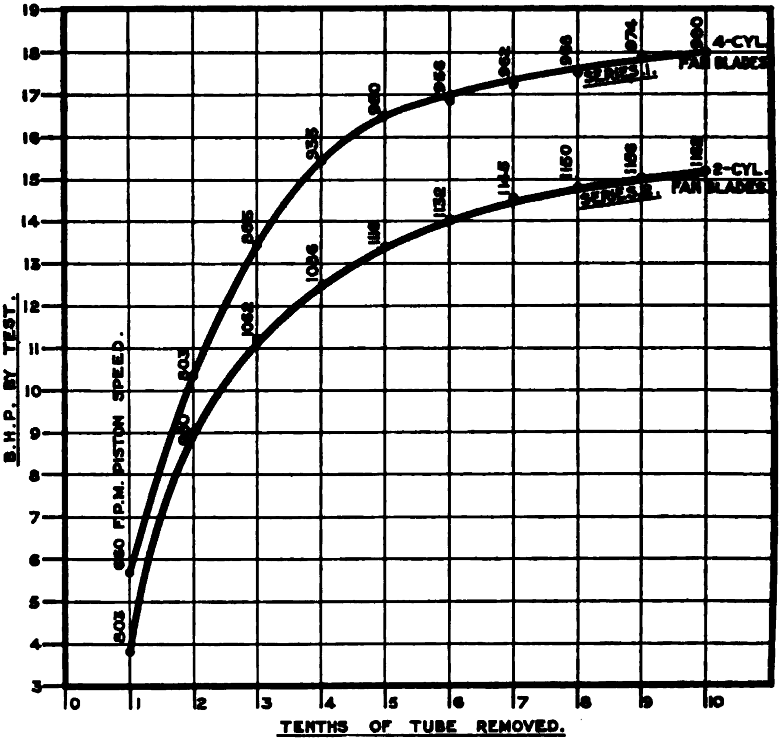

FIG. 342

feet per minute. Here $d = 3{\cdot}15$ ins., so that we have, v denoting the mean velocity in feet per minute across area A sq. ins., necessary to fully charge the cylinder:

$$v = 7{\cdot}8 \;.\; \frac{\sigma}{A} \text{ ft. per min.}$$

Values of v calculated from this result, for the maximum area available, are tabulated in the right-hand column of each series, and in the diagram, fig. 343, values of ηp are plotted against the 'tenths of tube removed.' Comparison of the table and this figure shows that the mean pressure attains the normal value for a cylinder of this bore (viz. about 80 lbs. per sq. in.) when v has roundly the value 9400 ft. per min. This is rather high, the average value in normal practice being from 7000 to 8000 f.p.m. only.

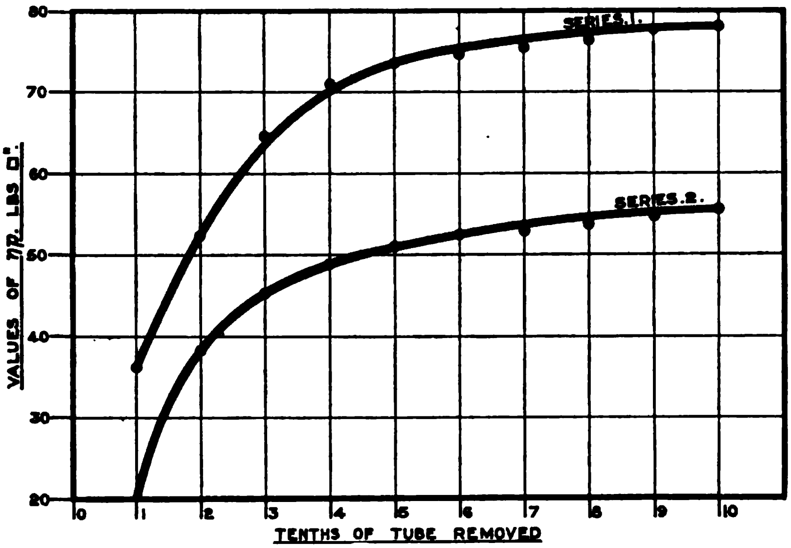

FIG. 343

These experimental results suggest that in designs where the valves are not placed in pockets, as, e.g. the Maudslay and Lanchester engines, the valve opening area $\pi\delta\lambda$ is more effective than in the usual arrangement, and hence that somewhat smaller valves may suffice in such designs.

Returning to the four-cylinder, 80 × 130 engine (figs. 340), it will be noted that the water spaces are capacious and also that the jacket only extends downwards as far as the lower edge of the piston when in its topmost position. Thermo-syphon cooling is adopted.

The inlet valves are 1·5 ins. diameter in the throat, with 0·34 in. lift; the exhaust valves are 1·34 ins. diameter, with 0·34 ins. lift; thus in this engine the inlets are greater in diameter than the exhausts.

The ratio of $\frac{\delta}{d}$ for the inlets is 0·477, and for the exhausts 0·426; the ratio of $\frac{\lambda}{\delta}$ for the inlets is 0·227, and for the exhausts 0·254.

The valve tappet rods are hollow and well guided, and the clearance between tappets and valve stems is capable of adjustment. Fibre buffers are fitted to prevent noise.

The same valve setting is used both for cast-iron and steel pistons. The inlets open 44° late and close 28° late; the very late inlet opening is noticeable in this engine. The crank turns through 35°, corresponding to nearly $\frac{7}{16}$ in. (11 mm.) of the down-stroke of the piston between the closing of the exhaust and opening of the inlet valve. This is to ensure that when the inlet opens the pressure in the cylinder is below that in the inlet pipe, thus avoiding resurgence and maintaining a high volumetric efficiency at high speeds.

The piston speed at maximum power (*v.* fig. 344) with cast-iron pistons is 1660 ft. per min.; the corresponding mean velocity through the valves *v*, in feet per minute, by Eq. (31), is:

$$\text{For the inlets:} \quad \left(\frac{3{\cdot}15}{1{\cdot}5}\right)^2 \times 1660 = 7300.$$

$$\text{For the exhausts:} \quad \left(\frac{3{\cdot}15}{1{\cdot}34}\right)^2 \times 1660 = 9200.$$

This type of engine is also fitted for racing purposes with steel pistons, thus reducing the mass of the reciprocating parts and permitting a higher piston speed to be maintained for racing purposes. To allow for the increased quantity of mixture then needed the valves are increased in diameter to 1·63 ins. and 1·47 ins. respectively, and a 45 mm. carburettor is then used (*v.* Chap. IX).

Fitted with light steel pistons and enlarged valves, the HP, σ graph is very straight up to a piston speed of 1900 ft. per min.; the power at this speed is 47·4, and still increasing with σ; the graph is shown in fig. 344.

At 1900 ft. per min. we have, for the mean velocity through the valves:

$$\text{Inlets:} \quad \left(\frac{3{\cdot}15}{1{\cdot}63}\right)^2 \times 1900 = 7100 \text{ ft. per min.}$$

$$\text{Exhausts:} \quad \left(\frac{3{\cdot}15}{1{\cdot}47}\right)^2 \times 1900 = 8700 \text{ ft. per min.}$$

These figures are in practical agreement with those obtained when cast-iron pistons are used.

The pistons illustrated in figs. 340 are of cast iron, flat-topped, with a length equal to their diameter only; each piston has four spring rings, the lowest serving to retain the gudgeon pin in place; a fifth 'scraper' or oil-excluding ring is fitted near the bottom of the piston. To assist the conduction of heat from the centre of the piston a series of three concentric annular cooling ribs will be noted on the under side.

As illustrated the design is very substantial, and could probably be somewhat reduced in weight without sacrifice of necessary strength.

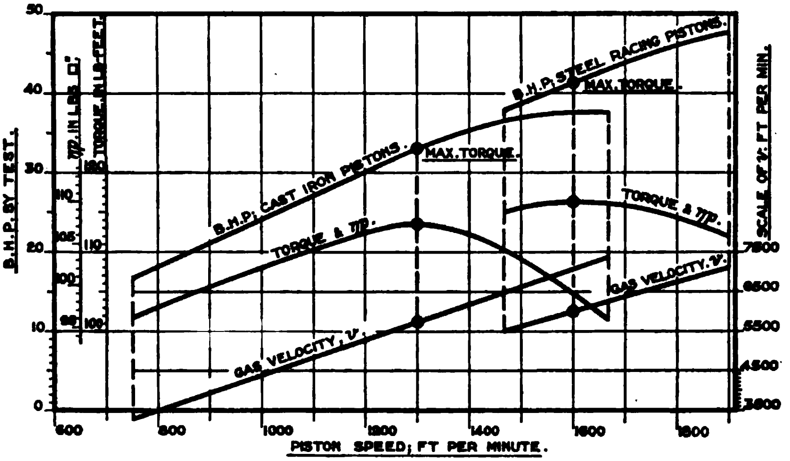

FIG. 344

The gudgeon pins are of solid steel, $\frac{1}{2}$ in. in diameter, convex-ended; the length of the gudgeon bearing is $1\frac{3}{4}$ ins.

At the maximum explosion pressure of, say, 300 lbs. per sq. in., the gudgeon is momentarily loaded with 2340 lbs., corresponding to $\frac{2340}{0{\cdot}5 \times 1{\cdot}75} = 2700$ lbs. per sq. in., roundly, of projected area of bearing.

The connecting-rods are stamped from 50–60 ton high carbon steel, and are of the usual H-section, $11\frac{3}{8}$ ins. between centres; the ratio of this to the crank-length is $\frac{11{\cdot}375}{2{\cdot}56} = 4{\cdot}45$, which is practically $\frac{9}{2}$. The unusual lightness of the 'big end' is noteworthy in this design;

the white metal big-end bearing is 1¼ ins. in diameter, and 1¾ ins. long; the maximum pressure per sq. in. of projected area is therefore

$$\frac{2340}{1{\cdot}25 \times 1{\cdot}75} = 1070 \text{ lbs. per sq. in. only.}$$

The piston and connecting-rod complete weigh jointly 4·65 lbs. with cast-iron pistons, and 3·96 lbs. when steel pistons are used.

The crankshaft is of nickel steel and, contrary to usual practice, is built up, the crank cheeks being prolonged to counter-weight the reciprocating masses. Messrs. White & Poppe only use solid crankshafts for their larger engines. The crank-pins are 1¼ ins. in diameter, and the shaft is 1⅛ ins. diameter; it is borne in five white metal bearings kept as close up as possible to the big ends. The bottom cover of the crank-chamber can be removed without disturbance of the main bearings.

The ignition is by high-tension magneto; a Bosch dual machine is employed (Chap. III); the sparking-plug is fitted in the inlet valve cap. Bosch *double* ignition can be fitted if desired, the second plug being then fitted in the exhaust valve cap. The ignition is hand-controlled. Sparking-plugs having substantial electrodes are favoured, as lessening the risk of pre-ignition at high compressions. The order of firing is 1342.

A White & Poppe 'No. 30'[1] variable jet carburettor is normally used. Mr. Poppe has experimented with a single jet, and air sleeves of various sizes, and finds that for best power results a constant relation should subsist between the area of jet and that of the air sleeve. Accordingly in the White & Poppe variable jet carburettor the air-regulating sleeve carries a central tube embracing the jet column, and having an eccentric hole at its upper end so arranged that both air area and effective jet area vary together, thus preserving the correct relation at all times. This carburettor is fully described and illustrated in Chap. IX of this volume.

The lubrication is semi-mechanical. Oil contained in an external reservoir is delivered by a pump in excess to the main bearings; the overflow from these fills the troughs under the big ends shown in figs. 340. Scoops on the big ends take up the oil for the crank-pin bearings, and the pistons and gudgeons are lubricated by the splash. The troughs are maintained in an overflowing condition, and the overflow collects in a sump and is thence returned to the external reservoir by a second pump. It will be noted that the gudgeons have small troughs on top for collection of lubricant.

The weight of the engine complete with carburettor and magneto, but without flywheel, is 372 lbs.; this gives the following figures per HP:

---

[1] The number expresses the bore in mm.

At normal piston speed of 984 ft. per min., 23½ BHP . . 15·8 lbs.
At maximum piston speed of 1660 ft. per min., 37½ BHP . 10·0 lbs.

with cast-iron pistons. With light steel pistons the maximum piston speed rises to 1900 ft. per min., and the BHP to 47½, corresponding to only 7·85 lbs. per BHP.

In fig. 344 the graph of the power and speed of this engine is given both with cast-iron and also with steel pistons; at the assumed normal piston speed of 984 ft. per min. the BHP is 23½; at the maximum (with cast-iron pistons) it is 37½; the ratio of normal to maximum is here therefore 0·63 only. The estimation of normal horse-power is, however, purely arbitrary; in general for touring car engines the value is from 0·8 to 0·9.

THE FOUR-CYLINDER, 3·15 INS. × 5·12 INS. WHITE & POPPE ENGINE

Piston speed, ft. per min.	Revs. per min.	BHP by test	Mean torque in lb.-feet	Value of ηp. lbs. per sq. in.	Notes
750	880	17·0	101·6	96·2	
900	1055	21·2	105·3	99·7	
1000	1175	24·1	018·0	102·2	
1100	1290	27·0	019·9	104	
1200	1410	30·0	112·0	106	
1300	1525	32·9	113·3	107·2	Max. torque with cast-iron pistons
1400	1640	35·0	112·0	106	
1500	1760	36·6	109·2	103·4	Cast-iron pistons
1500	1760	*38·7*	115·5	*109·3*	Steel pistons
1600	1875	37·5	105·0	99·3	Cast-iron pistons
1600	1875	*41·4*	116·0	*109·8*	Steel pistons
1660	1950	37·5	101·3	96	Max. HP with cast-iron pistons
1700	2000	*43·8*	115·5	*109·3*	Steel pistons
1800	2110	*46·2*	114·3	*108·8*	,, ,,
1900	2230	*47·5*	112·0	*106*	,, ,,

The test results from this engine are remarkable for the very high and constant value of the brake mean effective pressure ηp; reference to the graph and the accompanying table shows that from piston speeds of 900 to 1900 ft. per min. the value of ηp lies between the high and narrow limits 100 to 110 lbs. per sq. in. roundly. This is a noteworthy performance, resulting from the high compression, the large inlet valve diameter—which here exceeds that of the exhaust—and the large (45 mm.) W. & P. variable jet carburettor employed.

In another set of test results from a series of engines by the same makers with which the authors have been supplied, the maximum value of ηp occurred in that case in which, alone of the series, the inlet valve diameter was greater than that of the exhaust.

As hot gas flows more freely than cold gas, and as, moreover, the difference of pressure on the two sides of the valve available for producing flow is probably usually greater in the case of the exhaust valve than of the inlet, it appears to be a reasonable conclusion that an inlet valve diameter exceeding that of the exhaust is good practice.

The high and constant value of ηp involves also a high and constant value of the mean torque (see Eq. (39) of the preceding chapter). In the torque graph, fig. 344, the scale is very extended, but reference to the table shows that between piston speeds of 900 and 1900 ft. per min. the torque only varies between the limits 100 to 116 lbs.-ft., roundly. With cast-iron pistons the maximum torque here occurs at a piston speed of 1300 ft. per min., the BHP then being 32·9; this is considerably in excess of the 'normal' power of 23·5, and is 0·88 of the maximum power. Thus the maximum torque here occurs very far up the horse-power speed curve; it is very generally found at quite low piston speeds, and at much under the normal horse-power. The whole range in the torque variation is here, however, so very small that it may, roughly, be regarded as constant over the range of the test speeds; this implies that the power speed graph is approximately a straight line passing through the (0 0) origin of co-ordinates.

On a trial of a new engine of 50 consecutive hours' duration at the normal speed of 1150 r.p.m. the BHP developed was 23·5, and the petrol consumption per BHP hour 0·65 pint of 'Shell' spirit, having a specific gravity of 0·72 at 60° F., and a heat value of 16,900 B.Th.U. (° F.) per pint. 'Vacuum A' lubricating oil was used, and the consumption amounted to 0·029 pint per BHP hour. After six hours' running the oil temperature in the service reservoir was 122° F.; at the end of the fifty hours the temperature had fallen to 113° F., due to the settling of the shafts, &c., in their bearings. The power was absorbed by a carefully calibrated air fan dynamometer.

The fan vanes, two in number, are of steel plate about $\frac{1}{8}$ in. thick, and square in form; the distance from centre to centre of the vanes is kept about 3·6 ft. for all cases. Each pair of vanes is calibrated by direct comparison with an independent electrical test.

The brake thermal efficiency from the above test results is

$$\frac{2545}{0{\cdot}65 \times 16{,}900} = 0{\cdot}232.$$

The air standard efficiency corresponding to a volume ratio of compression 4·7 is 0·466 (*v.* Vol. I, p. 248).

Hence the brake relative efficiency is $\frac{0{\cdot}232}{0{\cdot}466} = 0{\cdot}5$.

Allowing that, on account of the working mixture being not ideal air of constant specific heat, but a mixture of gases of specific heat increasing with the temperature, the maximum theoretical efficiency is only 0·8 × 0·466 = 0·3728, we have as the absolute brake thermal efficiency $\frac{0{\cdot}232}{0{\cdot}3728} = 0{\cdot}623$ (*v.* Vol. I, p. 274).

The Four-Cylinder, 91 × 146 *mm. Vauxhall Engine.*—Longitudinal and transverse sections of this engine are shown in figs. 345 A and B.

The bore is 3·58 ins., stroke 5·75 ins.; the ratio of stroke to bore is 1·604. Rating for taxation purposes, 20·5.

The cylinders are cast 'en bloc'; the crankshaft being borne in five bearings, there is ample water space round each cylinder. The cylinder walls are $\frac{1}{4}$ in. thick; the combustion chamber head is domed, and about $\frac{5}{16}$ in. in thickness; the valves are placed side by side in a pocket on one side of the cylinder.

The volume ratio of compression, C, is 5·2; the corresponding compression pressure from Eq. (37) is 133 lbs. per sq. in. absolute. The volume of the combustion chamber is accordingly (Eq. 35) 13·8 cub. ins., while its total surface is about 70 sq. ins., of which 45 sq. ins. are water cooled and 25 are not.

The ratio of surface to volume in the combustion chamber is thus $\frac{70}{13{\cdot}8} = 5{\cdot}1$; a hemispherical chamber of the same cubical content would have a ratio of 2·4 only.

The total surface exposed to the working gases when the piston is at the bottom of its stroke is about 135 sq. ins.; hence the ratio of combustion chamber surface to total surface exposed is $\frac{70}{135} = 0{\cdot}52$.

The cylinders are jacketed nearly to the bottom of the piston stroke; cooling is on the thermo-syphon system.

The inlet and exhaust valves are each 2 ins. in diameter, with $\frac{1}{2}$ in. lift. Thus the ratio $\frac{\delta}{d}$ has here the large value 0·56, while $\frac{\lambda}{\delta}$ is 0·25. The tappet rods are hollow, and provision is made for adjusting the clearance. The valve spring load is 36 lbs. when shut, and 108 lbs. when open. The valve setting is as follows:

Inlets: open 20° late and close 20° late.
Exhausts: open 35° early and close 6° late.

Thus the inlets open 14° after the exhausts close, and regurgitation into the inlet pipe is avoided.

The piston speed at maximum power (*v.* fig. 346) has the very high value of 2110 ft. per min., the corresponding mean velocity v through the valve throats being (Eq. (31)) 6750 ft. per min., a satisfactorily low figure ; for many touring car engines the value is between 7000 and 8000. This engine is used for racing purposes.

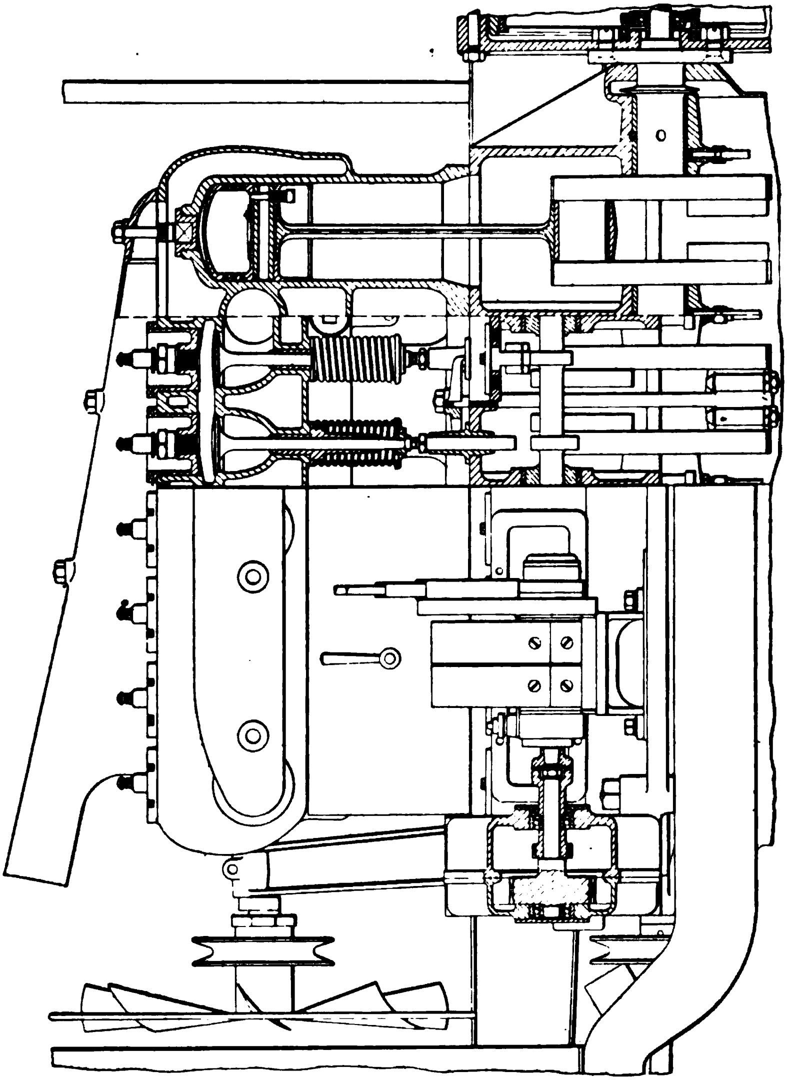

FIG. 345A

The pistons are of steel, with domed tops, very thin in the centre ; each piston has three spring rings. The weight of a piston complete with rings and gudgeon pin is only 2·5 lbs. The length is equal to the bore. The gudgeon pins are hollow, and $\frac{3}{4}$ in. in diameter ; the length of the gudgeon bearing is 2 ins.

At the maximum explosion pressure of roundly 300 lbs. per sq. in. the gudgeon is momentarily loaded with 3000 lbs., corresponding to

$$\frac{3000}{2 \times \frac{3}{4}} = 2000 \text{ lbs. per sq. in. of projected area.}$$

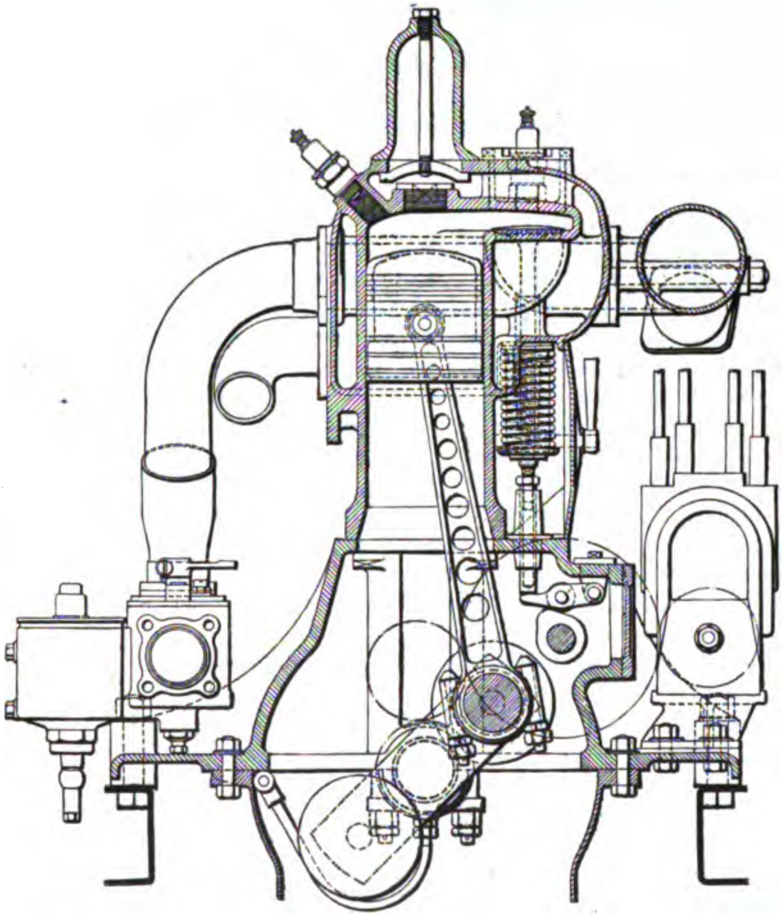

FIG. 345B

The connecting-rods are high-tension mild steel stampings of the usual H-section ; to reduce reciprocating weight nine holes of diameter varying from $\frac{7}{16}$ in. to $\frac{3}{4}$ in. are drilled through the web. The distance between centres is 11·6 ins.; the ratio of this to the crank length is $\frac{11·6}{2·88} = 4·03$; the usual value of this ratio in normal touring engines is 4·5, but with the very light reciprocating parts and longer strokes now met with the proportion is sometimes found to approach the value 4.

The big end is here unusually light; a similar feature exists in the 15·9 White & Poppe engine already described. The white metal big-end bearing is 1¾ ins. in diameter and 2¼ ins. long; the maximum pressure per sq. in. of projected area is therefore $\frac{3000}{1·75 \times 2·25}$ = about 765 lbs. per sq. in. only.

The weight of a connecting-rod complete, excluding gudgeon pin, is 4·13 lbs.; as the piston complete weighs 2·5 lbs., the total weight of the reciprocating parts per cylinder is 6·63 lbs.

The crankshaft is of vanadium chrome steel, borne in five white metal bearings; the crank pins are 1¾ ins. in diameter and 2¼ ins. long; the shaft diameter is also 1¾ ins. The bottom cover of the crank-chamber is removable without disturbing the main bearings.

The ignition is by high-tension magneto; the igniting plug is placed over the inlet valve; if double ignition be desired, a second plug may be placed over the exhaust; as shown in the plate there are three plugs in each cylinder; normally only that over the inlet is used. The plug over the exhaust is found to occasionally cause pre-ignition trouble, while the third (inclined) plug through the cylinder wall tends to become oil fouled.

The carburettor is a 40 mm. (1·58 ins.) White & Poppe, with variable jet.

There is forced lubrication to the main shaft bearings; thence the oil passes by ducts in the crank cheeks and pins to the big ends. The gudgeons and pistons are splash lubricated.

The weight of the engine, including the flywheel and all accessories rendering it a complete power unit in readiness to run when coupled up to an exhaust pipe, is 440 lbs. At maximum torque (fig. 346) the BHP is 57½; the corresponding weight per BHP is 7·65 lbs. At maximum power this figure is reduced to 7·35 lbs. per BHP; these very low figures result from the extremely high piston speed attained by this engine.

A power speed graph is given in fig. 346; graphs of the torque, ηp, and mean velocity of gas through valves are also exhibited; the tests were made with an air brake dynamometer of the same type as that used by Mr. Poppe.

In the accompanying table corresponding values of speed, power, torque, and ηp are given.

THE FOUR-CYLINDER, 3·58 INS. × 5·75 INS. VAUXHALL ENGINE

Piston speed, ft. per min.	Revs. per min	BHP by Test	Torque in lb.-ft.	ηp, lbs. per sq. in.
1092	1140	31·0	142·8	93·2
1250	1300	35·6	143·3	93·5
1500	1570	43·0	144·3	94·2
1750	1825	50·0	143·8	93·8
2000	2090	57·5	144·8	94·4
2100	2190	59·8	143·3	93·5
2250	2350	51·0	114·0	74·4

The power speed graph for this engine is extraordinarily straight, and its direction passes through the zero origin; this straightness results largely from the generous proportions of the valves and piping; the gas velocity even at the extremely high piston speed of 2110 ft. per min. has been seen to be only 6750 ft. per min. The cylinders accordingly obtain a charge of normal amount at every suction stroke, the brake mean effective pressure and torque remain sensibly constant, and the BHP increases in direct proportion to the speed.

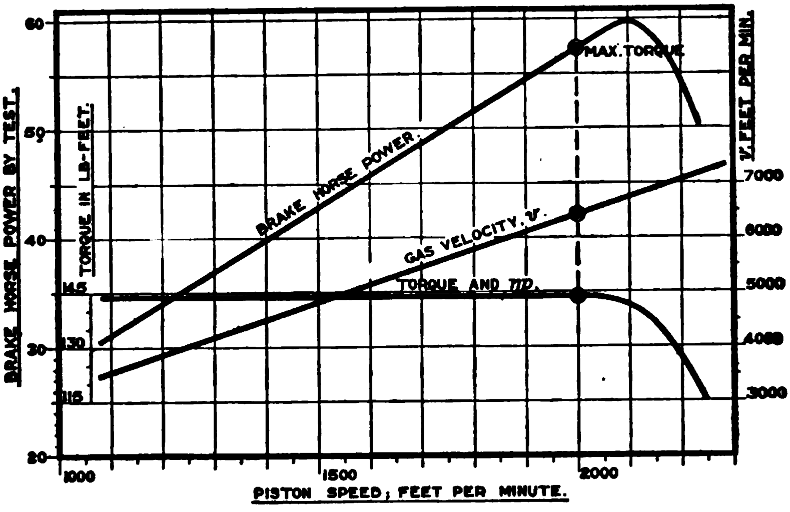

FIG. 346

The mechanical efficiency η diminishes with increase of speed (*v.* Chap. VII), while the mean effective pressure p increases with speed—in the absence of wire-drawing—agreeably with Eq. (39F) of Chap. IX; in this engine the changes in η and p just neutralise one another, the product ηp remaining constant up to the very high piston speed of 2100 ft. per min.

The petrol consumption at full power has been found to be roundly 0·7 lb. per BHP hour; this corresponds to a heat supply of $\frac{0\cdot 7 \times 18{,}600}{60} = 217$ B.Th.U. per BHP minute, i.e. to a thermal efficiency of $\frac{42\cdot 4}{217} = 0\cdot 196$.

The air standard efficiency corresponding to a compression ratio of 5·2 is 0·49; the relative efficiency is thus 0·40.

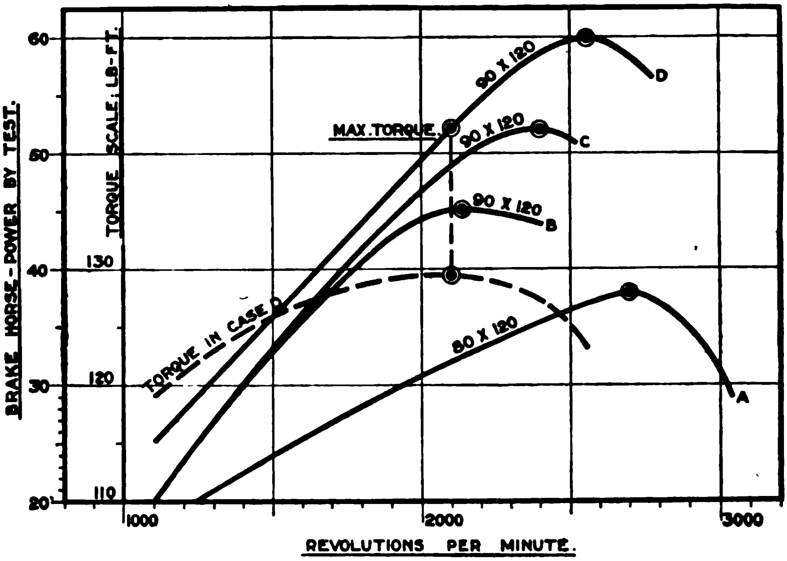

FIG. 347

Assuming as before that the highest attainable efficiency is 0·8 of the air standard, the true relative efficiency is expressed by 0·5.

Some results of further experiments made by the Vauxhall Company during 1910 on their four-cylinder, 90 × 120 mm. (3·54 ins. × 4·73 ins.) engine may be conveniently given here.

Fig. 347 shows four power-speed graphs; that marked A was obtained from an engine with cylinders bored out to 80 mm. (3·15 ins.) only. A maximum of 38 BHP occurred at 2700 revolutions per minute. The compression pressure was, however, deemed to be too low; measured by an Okill gauge the compression pressure was 82 lbs. per sq. in. Curves B, C, and D refer to an engine 90 × 120; the same engine was used in all three cases; the compression pressure, by an

Okill gauge at 1000 r.p.m., had here th
each case.

The improvements in speed and pow
and D were achieved partly by reduct
ciprocating parts, partly by carburettor
partly by improved valve setting ; the d
inferred from the following figures :

Case	M
B	
C	
D	

In the appended table some further
the valves were $1\frac{3}{4}$ ins. in diameter with
of gas at different engine speeds has been
It will be noted that the values of ηp i
sq. in. between 1600 and 2400 r.p.m., and r
over that range the torque is correspondin
—so far as one exists—occurs at the ver
revolutions per minute, and at 0·87 of
mean velocity of gas through the valve h
roundly 6800 ft. per min.

THE VAUXHALL 90 × 120 : EXPER

Speed.		BHP by test.			F
Revs. per min.	Piston speed, f.p.m.	B	C	D	ηp in lbs. per sq. in.
1100	870	20.0	20·0	25·0	96·5
1600	1260	35·3	36·2	38·6	103·0
2000	1580	44·3	46·8	49·4	105·0
2150	1700	45·0	—	53·2	105·0
2400	1890	44·3	52·0	58·3	103·5
2550	2010	—	—	60·0	100·0

The Four-Cylinder, 25 *HP New Daimler*
engine with valves of the sliding sleeve type
in longitudinal and transverse section in fi
The bore is 3·94 ins. ; stroke 5·12 ins. ;
Rating for taxation purposes, 24·8.

The cylinders are cast in pairs, the dist
of the members of each being about 5·9 i
borne in five bearings there is an ample water
The working barrel of the cylinder is th

cating cast-iron sleeves controlling the periods of inlet and exhaust;

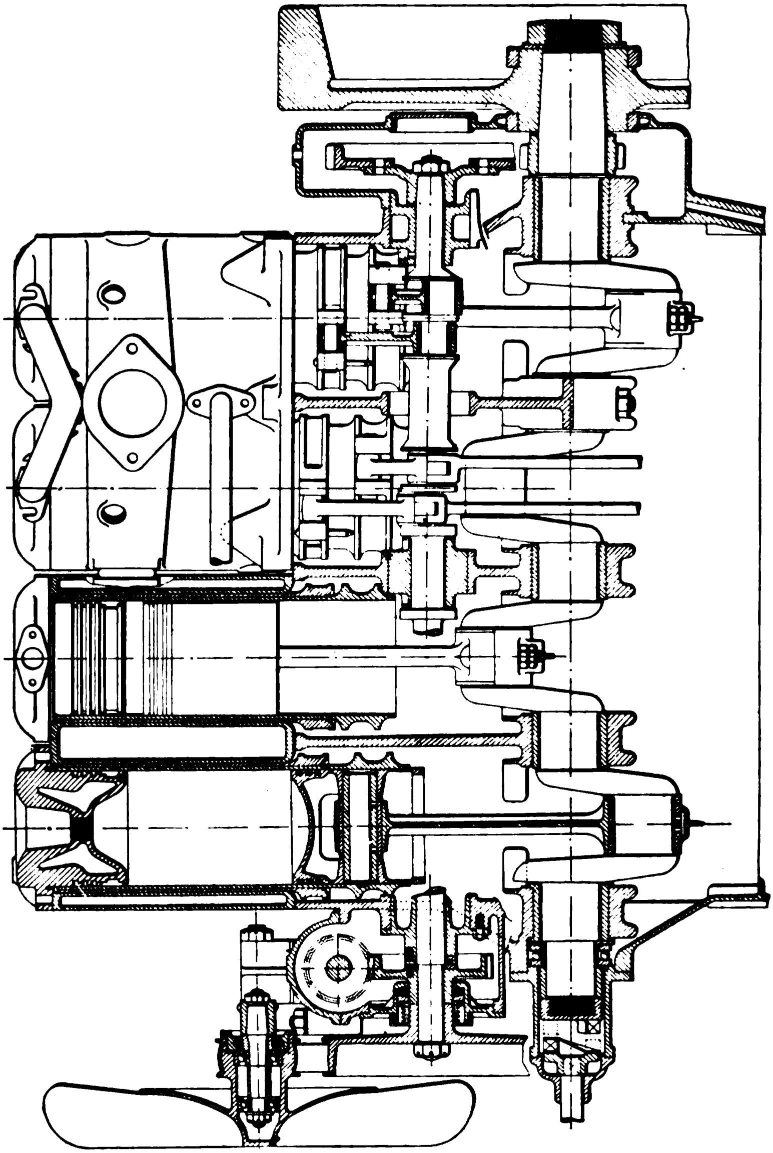

Fig. 348A

this and the outer sleeve within which it works are but about $\frac{1}{8}$ in. in thickness; outside these two sleeves is the cylinder proper, also

with walls only about $\frac{1}{8}$ in. in thickne
ground up dead true after machining.
between the working gases and cooling
in the barrel.

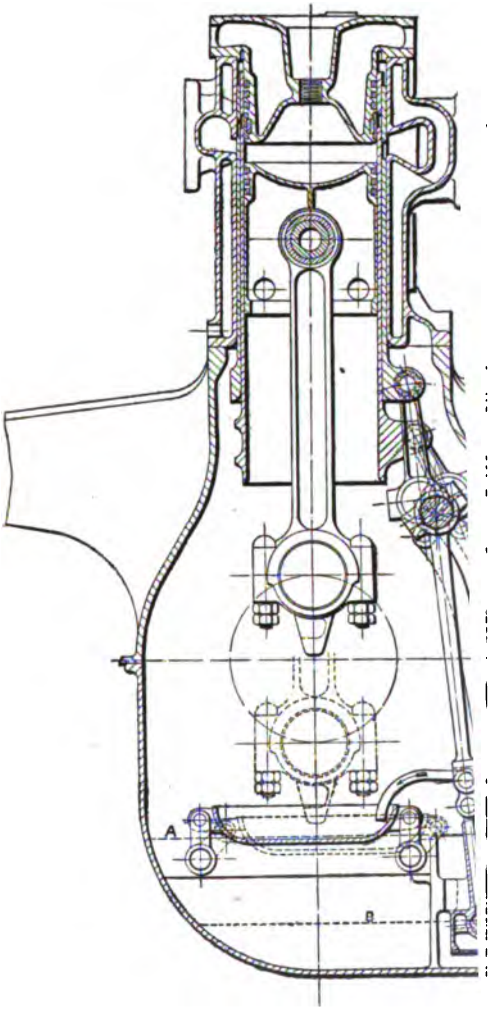

FIG. 348B

The sliding valve design necessitates
ends are closed by water-cooled heads,
$\frac{3}{16}$ in. thick only; each head is fitted wit
against the inner surface of the inner sleev

the lowest ring is 0·8 in. deep, as the ports in the inner sleeve overrun it in the course of their motion.

The open-ended design of cylinder enables an exceedingly compact combustion chamber to be obtained, and every part of the surface of the chamber, including the dished piston crown, is machined and polished, thus diminishing heat loss and retarding the deposition of carbon. The form of the chamber approximates more nearly to the spherical than in engines with poppet valves as usually designed.

The volume ratio of compression c is 5·1, corresponding to an adiabatic compression pressure of 122 lbs. per sq. in. (absolute).

In a six-cylinder, 3·94 ins. × 3·94 ins. engine of this type constructed by the Daimler Company the compression ratio was 5·35, corresponding to a pressure of 130 lbs. per sq. in.; signs of pre-ignition were observed in service. In the case of a similar smaller engine, 2·75 ins. × 4·5 ins., the compression ratio was 6·0, corresponding to a pressure of 152 lbs. per sq. in. absolute; this engine developed pre-ignition knock at full load.

Thus it appears that for regular every-day service the compression ratio should not exceed, roundly, the value 5.

By Eq. (35), *ante*, the volume of the combustion chamber is 15·2 cub. ins., while its total surface is estimated as 40 sq. ins.; the ratio of surface to volume has here, therefore, the small value $\frac{40}{15\cdot2} = 2\cdot63$ only. A hemispherical chamber of the same cubical content would have the value 2·33.

The total surface exposed to the working gases when the piston is at the bottom of its stroke is 104 sq. ins.; the ratio of combustion chamber surface to total surface exposed is thus $\frac{40}{104} = 0\cdot385$ only.

The cylinders are jacketed over the whole length of the working barrel; the 'en bloc' arrangement of four cylinders is not used in these engines on account of the distortion in working causing side wear and knock to soon occur in the first and last cylinders; they are accordingly cast singly or in pairs.

The valves consist of ports cut in the sliding sleeves within which the pistons work; these ports are clearly indicated in the transverse section (fig. 348B).

Each sleeve carries a pair of lugs and knuckle pin at its lower end, by which it is caused to reciprocate by means of a short connecting-rod driven from the timing shaft; these also are clearly shown in the transverse view. The stroke of each sleeve is about ⅞ in.

Each inner sleeve is estimated to weigh about 6¾ lbs., and each outer sleeve about 5½ lbs.; hence at the normal crankshaft speed of 1200 r.p.m. the maximum force of acceleration for the inner sleeve is by

Eq. (6), *ante*, 36 lbs. weight, indepen
ances. The acceleration force is thus
the large bearing surface between the
looking, one-sided drive adopted to o
running.

The maximum revolution speed of
2800 r.p.m.; the acceleration force for
a maximum value of roundly 200 lbs.

The maximum inlet area is about 0·41
the maximum exhaust area is 0·435 in. ×

The value setting is as follows:

Inlet: opens 0° late and
Exhaust: opens 60° early an

There is thus an overlap of 18° in the v
of the exhaust prevents whistling of the i
to involve no disadvantage; its early
small sacrifice of power; 60° early corresp
in this case. The late inlet closing has
experience; it enables the best volumetr
by the engine at high revolution speeds.

Shallow grooves are formed in the o
and in the larger engines the surface i
exhaust port is also slightly recessed by
after a short period of running these
become filled with a deposit of carbon, w
and assists lubrication.

An area of inlet port of 1·3 sq. ins. is
valve of 1·29 ins. diameter in the throat;
valve for an engine of this size, good prac
about 1⅝ ins. The mean velocity of gas thr
high in this engine (see table on p. 610
min. at the low piston speed of 750 ft. per
of 11,250 ft. per min. at a piston speed of
effect of this is to cause the values of ηp a
decline with increase of speed, as indicate
torque occurs at a piston speed less t
normal running speed is 1200 r.p.m., a
developed.

The pistons are of cast iron, the cro
spherical segments of 2¾ ins. radius; each p
the length of piston is 1·15 times its diamet
and cooling web is cast on the under side,
and connecting this with the gudgeon bos

Speed		BHP by test.	ηp in lbs. per sq. in.	Torque in lb.-feet	Gas velocity, v, ft. per min.
Revs. per min.	Piston, ft. per min.				
880	750	27·5	99·4	164	7000
1055	900	32·5	97·8	162	8400
1175	1000	36·0	97·5	161	9350
1290	1100	39·0	96·0	159	10,300
1410	1200	42·0	94·8	157	11,250

A stiffening bead is left round the bottom, and the lower part of the piston is drilled with $\frac{3}{8}$ in. holes to help in preventing oil from passing up into the combustion chamber. With this design of piston it is found necessary to leave a small clearance to the surface in the neighbourhood of the ends of the gudgeon pin. The weight of a piston complete with rings and gudgeon pin is 3·8 lbs.

The gudgeon pin is of hardened steel and has the large diameter of $1\frac{3}{16}$ ins.; it is hollow, an $\frac{11}{16}$ in. hole being bored through it. The length of the gudgeon bearing is 2 ins. Assuming a maximum explosion pressure of roundly 300 lbs. per sq. in., the load due to this on the gudgeon per sq. in. of projected area is 1550 lbs. only.

The connecting-rods are nickel steel stampings of the usual H-section, 10 ins. long between centres; the ratio of this to the crank length is 3·9 only; a very usual value is 4·5.

The gudgeon end bush is of hardened steel; the big end is of bronze lined with white metal; this bearing is 2 ins. in diameter and 2 ins. long; the pressure per sq. in. of projected area, due to the explosion pressure of say 300 lbs. per sq. in., is only about 900 lbs.

The weight of one connecting-rod complete is 2·44 lbs.; as the complete piston weighs 3·8 lbs., the total weight of the reciprocating parts per cylinder is 6·24 lbs., a low figure.

The crankshaft is of pressed nickel chrome steel, borne in five white metal lined bronze bearings. It is of very substantial size, being 2 ins. in diameter in the journals and crank-pins. The rearmost bearing is $2\frac{7}{8}$ ins. in length, the other four being each 2 ins.; the total length of bearings is thus $10\frac{7}{8}$ in. The crankshaft is hollow, a $1\frac{3}{16}$ ins. diameter hole being bored through it; the disposition of the cranks is normal, viz. the two outer together and at 180° to the two inner, also together; the order of firing is 1243.

A ball thrust bearing fitted at the forward end of the engine resists any end pressure that may come upon the crankshaft.

In all petrol engines at certain speeds the working impulses synchronise with the natural vibration period of the crankshaft, and at such speeds large torsional oscillations occur, causing a 'thrashing'

noise and much increased engine vibra
new Daimler engines by a patented d
consisting essentially of a small flywhee
crankshaft, and driven through fluid
tions of the shaft are stated to be effe
claimed that with this adjunct it is unne
of a six-cylinder larger in diameter th
four-cylinder engine.

In the opinion of some designers,
from the rear end of the crankshaft als
oscillation.

The cast-iron flywheel is 20¼ ins. in
3⅝ ins.; the internal cone of the clutch is
of the rim at the normal speed of 1200 r
The wheel is attached to the crankshaf
key, and nut. The weight of the rim is

The half-speed shaft is driven by a
from the flywheel end of the crankshaft.
applications has given much trouble
here of large size in proportion to its wor
stretch occurs in service.

High-tension ignition is used, the Bos
adopted; the ignition is hand-controlle
in the centre of the water-cooled cylinder
through a leather disc coupling about ¼
of carburettor is fitted; for starting pu
opened, when a spray nozzle giving a spe
operation; the engine having started, a
out the first jet and brings into actio
mixture. With this engine a somewhat
best, possibly on account of the locatio
control is by throttle.

In the two-, four-, and six-cylinder en
the air for the carburettor is drawn fro
gives a quiet suction, keeps the air in
supplies a warm and oil-misty air to the
vaporisation of the petrol and lubricati
extra air inlet valve is fitted below the w
which contains the cylindrical throttle v
fed to the carburettor, a pressure of abo
to give an adequate supply. The inlet
the exhaust about 2¼ ins.; the exhaust

Semi-splash lubrication is used; an
plungers and a rocking valve (see tran

the half-speed shaft as shown, delivers oil in excess from the crank-chamber sump into each of the four troughs beneath the big ends; thence the engine is lubricated by the big-end splash. The fifth pump delivers to the silent chain drive of the half-speed shaft, to the skew gear drive of the magneto and water pump at the forward end of the engine, and to an oil circulation indicator on the dash-board.

The sump contains about 2½ gallons of oil, the level then being A C (transverse section); C is a cock for determining the correct level. The lowest level at which the pump can work is at B. After lubricating the engine the oil drains back, by way of the internal surfaces of the crank-chamber, through a filtering gauze, into the sump; as there is no external reservoir in this system the oil tends to get somewhat heated and smoky during prolonged hard running.

The big-end troughs are so connected with the throttle control as to be raised when the throttle is opened, thus providing increased lubrication at increased engine speed.

The circulation of the cooling water is maintained by a centrifugal pump; the circulating system is diagrammatically shown in the accompanying fig. 349; the cylinders and cylinder heads are connected in parallel. B B, B B, are baffles to prevent the water from short-circuiting direct across from the inlet to outlet orifices. The separate cylinder heads involve more piping connections and render the thermo-syphon system of cooling inapplicable to this type of engine; about four gallons of water are required for cooling purposes.

The radiator consists of sixty-four much flattened brass tubes placed in parallel, vertically, and with their long diameters in a 'fore-and-aft' direction. These tubes connect an upper reservoir of cast aluminium having the characteristic Daimler cooling ribs on top, with a lower copper reservoir about 2½ ins. in depth. Each tube is about 20 ins. in length, 3 ins. in depth, and 0·08 in. in (internal) width; the total cooling surface is about 60 sq. ft. A belt-driven fan carried on ball bearings maintains the necessary current of air through the radiator (*v.* fig. 349).

In fig. 350 the power graph is shown between piston speeds of 750 and 1225 ft. per min.; it will be noted that the torque steadily declines with increase of speed owing to the high gas velocity; the value of ηp declines also, in proportion to the torque, from about 99½ lbs. per sq. in. at the lower, to about 95 lbs. per sq. in. at the upper, limit of speed.

No petrol consumption test results have been obtainable by the authors, and hence no estimates of efficiency can be made in this case.

The Six-Cylinder, 38 HP Lanchester Engine.—Longitudinal and transverse sections of this engine are given in fig. 351.

The cylinder bore is 4 ins.; stroke 4 ins.; stroke-bore ratio 1·0;

the stroke is here smaller in relation
The rating for purposes of taxation is
The cylinders, here shown in pairs, a

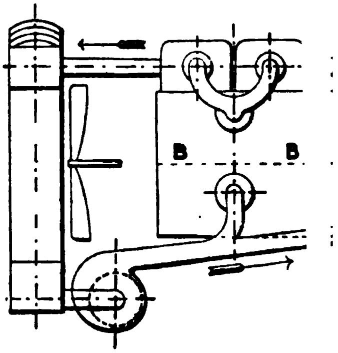

Fig. 349

of cold blast iron as hard as it is practicab
true by grinding.

As the crankshaft is borne in seven
water space around each cylinder, and t

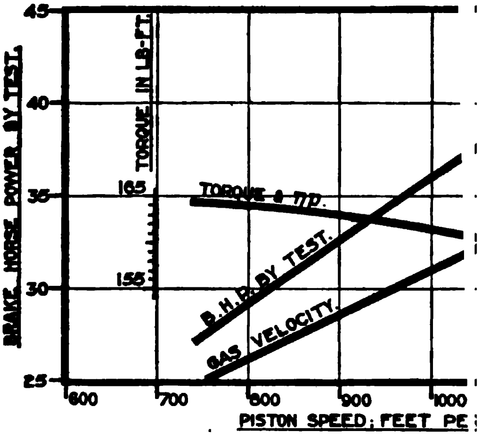

Fig. 350

especially well cooled ; the jacket extend
working barrel, and the water spaces of
communicate through holes 2½ ins. in diam
a stout rubber ring, as shown on p. 615.

The combustion chambers are of unus

Lanchester's use of horizontally placed valves; the arrangement possesses the important advantages of giving a simple, compact, and well-cooled form of chamber without corners or passages, and great facility of access to the valves and their seats; the valve disposition also ensures that the whole periphery is effective for the inflow and outflow of gas.

The volume ratio of compression c is 4·7, corresponding to a compression pressure by Eq. (37) of about 116 lbs. per sq. in. absolute; it is stated that with ordinary valve setting about 111 lbs. per sq. in. absolute is actually realised at normal speed. Corresponding to a compression ratio of 4·7 the volume of the combustion chamber is (Eq. 35) 13·6 cub. ins., its total surface being about 46 sq. ins., 29 of which are water cooled and the remaining 17 not; the ratio of surface to volume is thus $\frac{46}{13\cdot 6} = 3\cdot 38$; a hemispherical chamber of the same cubical content would have a surface volume ratio of 2·41 only.

The total surface exposed to the working gases when the piston is at the bottom of its stroke is, roundly, 96 sq. ins.; hence the ratio of combustion chamber surface to total surface exposed is $\frac{46}{96} = 0\cdot 48$.

The inlet and exhaust valves have each a diameter of 1·56 ins. in the throat, with a lift of 0·35 in.; the diameter is in accord with Eq. (32A) of Chap. VII. The value of $\frac{\delta}{d}$ is 0·39, and of $\frac{\lambda}{\delta}$ is 0·224. Both valves are cone-seated and connected with the stems by fillets of large radius; the stems are unusually stout, their diameter being no less than $\frac{9}{16}$ in.

In Messrs. Lanchester's four-cylinder engines there is frequently an overlapping valve setting, the inlets opening shortly before the exhausts close; in this six-cylinder type there is, however, no overlap, the setting being as follows:

Inlets:	open 7° late	close 20° late.
Exhausts:	open 40° early	close 7° late.

The neat method by which the valves are mechanically operated is clearly shown in the section, fig. 351; the two camshafts are situated about mid-way up the cylinders, and are driven by the usual half-speed gear through an idle wheel from the rear end of the crankshaft; the cams actuate the lower ends of the light rocking levers shown, the upper ends being in contact with the valve stems. The valves are held to their seats by light broad-based plate springs clearly shown in the illustrations; the use of flat springs saves a little weight, gives a rapid return to the valves, and is convenient in this arrangement of the valves; the springs as fitted do not become heated, and are very easily replaced in the event of breakage. The springs are connected with the valve stems by the small shackles indicated.

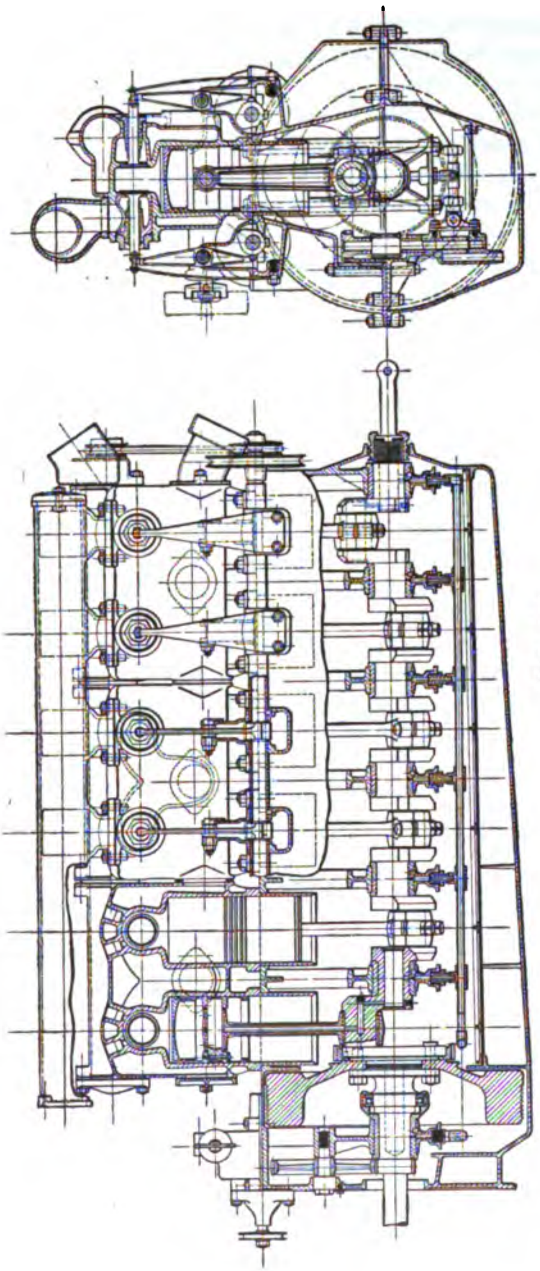

FIG. 351

The cams revolve in small pockets formed in the crank-chamber casting; these pockets form oil reservoirs wherein the cams and lower ends of the rocking levers work immersed.

The inlet valve with its seating casting is detachable, and may be removed from the cylinder by unscrewing the hexagonal nuts shown; the exhaust valve seating is formed in the cylinder casting, and the valve itself is inserted and removed through the hole into which the inlet seating casting is fitted; it will be seen that the exhaust valve seat and stem are adequately water cooled.

Maximum power is attained at the comparatively low piston speed of 1465 ft. per min. (see fig. 353), corresponding to a mean velocity v through the valve throats of 9600 ft. per min. by Eq. (31) of Chap. VII. This is a rather high value, the usual figure for engines with the ordinary valve arrangement being about 7000 ft. per min. only; it will be noted that the power graph, fig. 353, soons falls away from the maximum torque line as the speed is increased; greater power could probably be obtained from this engine if desired by increasing the valve throat diameters.

The flat-topped pistons are of pressed mild steel machined all over; each contains four cast-iron spring rings above the gudgeon and a fifth oil-excluding ring near the bottom; the rings are fitted with snugs to prevent them from so turning as to bring their split ends into line. The gudgeon pin is of solid steel $\frac{7}{8}$ in. in diameter, and is secured by a single locked set screw in one of the piston bosses as shown. The overall length of each piston is $4\frac{9}{16}$ ins., i.e. 1·11 times the bore, and the weight complete with rings and gudgeon pin is 3·93 lbs.

The length of the gudgeon bearing is $2\frac{1}{2}$ ins.; at the maximum explosion pressure of roundly 300 lbs. per sq. in. this bearing is momentarily loaded with 3770 lbs., corresponding to $\frac{3770}{2{\cdot}5 \times 0{\cdot}857} = 1725$ lbs. per sq. in. of projected area, a normal figure.

The connecting-rods are nickel steel stampings, the material having an elastic limit between 30 and 40 tons per sq. in.; the cross section shown in fig. 352 is of the usual **H**-type, but with a duct along the axis whereby oil is supplied to the gudgeon bearing from the crank-pin. The length between centres is $8\frac{3}{4}$ ins., the ratio of this to the crank length being 4·38, a normal proportion.

The weight of each rod complete is 4·0 lbs., almost exactly equal to that of the piston; the total weight of the reciprocating parts is thus 7·93 lbs., which is rather great, especially as pressed steel pistons are employed; see Chap. VII.

The big ends are white metal lined, the alloy being bedded direct into the steel end of the rod; a light bearing is thus obtained; the gudgeon eye is bronze bushed.

The big ends are 1⅞ ins. in dia
about 1⅞ ins. of length is, however
fillets; hence the maximum pressure
of bearing is

$$\frac{3770}{1·875 \times 1·875} = 1070 \text{ lbs}$$

The crankshaft is in one piece and i
steel; it is carried by seven bronze bea
the shaft is 1⅞ ins. in diameter with a
right through it axially; end closers a
as the oil for the lubrication of the cra
and gudgeons is conveyed by way of th
the arrangement of ducts is clearly show
longitudinal section, fig. 351.

Each main bearing is 2⅛ ins. in leng
the exception of that beyond the f
which has a length of 2¾ ins.; the tota
of bearing is accordingly 15½ ins.; the b
are unaffected by the removal of th
cover of the crank-chamber.

The flywheel is enclosed within an ex
of the crank-chamber, and is bolted
spigoted flange-cheek of the sixth cra
shown; contrary to usual practice, it
overhung, the seventh bearing being b
this seventh bearing also houses a ball th
prevent any racking of the crankshaft.

The flywheel has an outside diameter
ins., a face width of 3¼ ins., and thickn
rim 2¼ ins.; the weight of the rim is acc
energy at the normal full speed of 1400 re
8600 ft.-lbs.; the complete flywheel weig

The carburettor is of the well-known I
and is illustrated and described in Cha
total weight is 29½ lbs.; but this figure i
mechanisms, &c., formed on the carbure
weight may be considered as about 24 lbs.

There is forced lubrication to all the
the gudgeons; a gear pump driven from
from the crank-chamber sump into the lo
shown in fig. 351, whence it reaches the
some passes within the crankshaft to the
ducts, and so to the big-end bearings; th
in the connecting-rods, to the gudgeons.

The outflow from all the bearings gra

the pump maintains a continuous circulation. The cylinders are lubricated by the surplus oil whirled from the crank cheeks; the cams and cam shafts are externally lubricated, the cams, as already mentioned, working immersed in oil baths.

Thermo-syphon cooling is employed, and the large water spaces and 2½ ins. bore water connections will be noted. The radiator is built up of thin vertical cellular tubes in parallel, connecting a top and a bottom tank; the total tube surface is 98 sq. ft.

The ignition is hand-controlled high-tension magneto, a Bosch dual machine being employed (*v.* Chap. III); the igniting plugs are located in pockets in the tops of the combustion chambers as shown in fig. 351; the firing of the mixture in the pocket causes a flash of flame to be projected into the combustion chamber, and is considered to produce crisp ignition; this device, first used by Daimler, has also been employed by de Dion and others.

It will be noted that the plug bosses are well water cooled, and that double ignition can be used if desired.

The crank throws are on the 'opposed three-cylinder' arrangement, viz. Nos. 1 and 6, 2 and 5, and 3 and 4 being severally together, and the order of firing is 153624.

The weight of the engine alone, including only the magneto, is 640 lbs., i.e. 16·8 lbs. per BHP at normal output. In addition there are the following items:

Flywheel	93 lbs.
Radiator	90 ,,
Fans	6½ ,,
Exhaust box	19 ,,
Carburettor	29½ ,,
Total	238 lbs.

Thus the total, complete with all accessories, is 878 lbs., or 23·1 lbs. per BHP of normal output. The maximum power is roundly 55½, corresponding to 15·8 lbs. per BHP of the complete engine.

The power speed graph is shown in fig. 353, together with curves of torque, ηp, and gas velocity, and the accompanying table exhibits numerical values of the torque and ηp over the range of speeds tested.

n, revs. per min.	σ, ft. per min.	BHP by test	Torque in lb.-ft.	ηp, lbs. per sq. in.	Notes.
660	440	24·2	192·5	96·2	
900	600	33·6	196·0	98·0	Max. torque.
1200	800	43·0	188·5	94·2	
1500	1000	49·6	173·5	86·8	
1800	1200	53·6	156·5	78·2	
2200	1467	55·5	132·6	66·3	Max. power

CHAPTER IX

ON CARBURETTORS

By far the greater part of the working fluid supplied to the internal combustion engine is air, which may be regarded in this connection as a mechanical mixture of oxygen and nitrogen roundly in the proportions of 23 per cent. of oxygen to 77 per cent. of nitrogen by weight; or 21 per cent. oxygen to 79 per cent. nitrogen by volume. Thus 1 lb. of oxygen is contained in 4·53 lbs. of air; and 1 cub. ft. of oxygen in 4·8 cub. ft. of air.

At atmospheric pressure of 14·7 lbs. per sq. in. (2116·8 lbs. per sq. ft.) and at a temperature of 32° F. (492° F. abs.), one pound of air occupies a volume of 12·36 cub. ft. At atmospheric pressure and 60° F. one pound of air occupies 13·07 cub. ft.

If p denote the pressure in lbs. per sq. ft. abs.; v the volume of 1 lb. in cub. ft.; and T the absolute temperature in ° F. (= ordinary temperature + 460); then p, v, and T for air are always inter-related in accordance with the equation:

$$pv = 53{\cdot}2\ \mathrm{T} \qquad (1)$$

A general account of the hydrocarbons included under the term 'petrol' is given in Chap. VI of this volume; it is sufficient to state here that in the majority of cases it has a specific gravity of about 0·72 at 60° F., so that the contents of an unopened two-gallon tin should weigh $2 \times 7{\cdot}2 = 14\frac{1}{2}$ lbs. roundly.

Its vapour density relatively to air is about $3\frac{1}{4}$; thus at atmospheric pressure and 60° F. one pound of petrol vapour occupies about $\frac{13{\cdot}07}{3{\cdot}25} = 4$ cub. ft. One pound of liquid petrol at atmospheric pressure and 60° F. occupies about $38\frac{1}{2}$ cub. ins. (0·0223 cub. ft.); hence at this pressure and temperature when converted into vapour it increases its volume by $\frac{4}{0{\cdot}0223} = 180$ times.

Laboratory experiments show that one pound of petrol vapour with from about 7 to 26 lbs. of air gives an inflammable mixture.

In engines working on the Otto cycle the fresh charge is, however, always diluted with some of the products of combustion of the preceding charge, and accordingly the weakest inflammable mixture in engines is about 20 lbs. of air and exhaust gases to one of petrol; for regular and satisfactory running mixtures containing less than 17·5 lbs. of air to one of petrol must be used. Practically the range of mixture in car engines is from about 11 to 17 lbs. of air to one pound of petrol.

If a mixture of air and coal gas be exploded, and the products of combustion be reduced to the original temperature and pressure, it will be found that a small *contraction* of volume has occurred (*v*. Vol. I, p. 331);

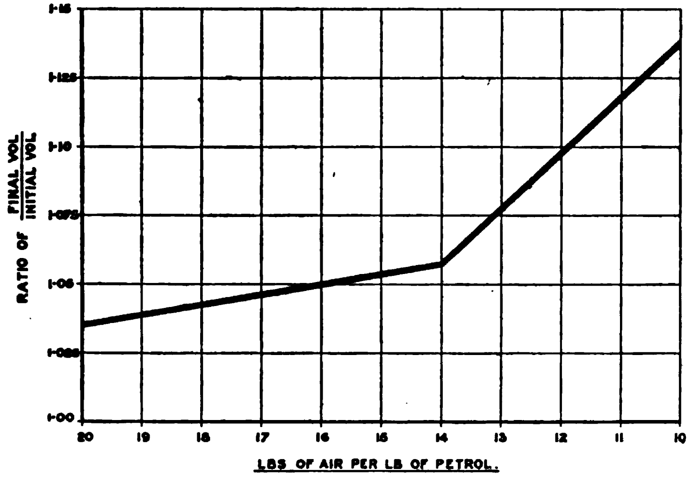

FIG. 354

with mixtures of air and petrol vapour, however, the reverse is the case, a small *expansion* of volume resulting with ordinary mixtures, while with strong mixtures the volume increase is quite large. In fig. 354[1] the value of the ratio $\frac{\text{Final volume}}{\text{Initial volume}}$ is plotted for different strengths of mixture; it will be seen that within a working range of about 13 to 17 the increase is small; but with the rich mixture of 10 to 1, the final volume is nearly 1·14 times the initial volume, whence the increased mean pressures obtainable by the use of rich mixtures. Thus, taking the three ratios of $\frac{20}{1}$, $\frac{14}{1}$, and $\frac{10}{1}$, we have the following as the approximate volumes of mixture in cub. ft. per lb. of petrol used, before and after combustion, at atmospheric pressure and 60° F.:

---

[1] Dr. W. Watson, *Cantor Lectures*, 1910, p. 26.

	Cub. ft.	Cub. ft.	Cub. ft.
Ratio . . .	$\frac{20}{1}$	$\frac{14}{1}$	$\frac{10}{1}$
Before . . .	265½	187	135
After . . .	275	197½	153

An average consumption for the petrol engines of cars on bench tests is about 0·65 lb. of petrol per BHP hour. With an average strength of mixture of about 15 to 1 by weight, giving about 1¾ per cent. of free oxygen in the exhaust so as to prevent the emission of any CO, this corresponds to $(15 \times 13{\cdot}07 + 4) \times 0{\cdot}65 \div 60 = 2{\cdot}17$ cub. ft. of mixture per BHP per minute at atmospheric pressure and 60° F. This number of roundly 2·2 cub. ft. per BHP minute is useful to remember for purposes of design.

Pratt's motor spirit has an average composition by weight of about 85·2 per cent. carbon and 14·8 per cent. hydrogen; hence for complete combustion to CO_2 and H_2O it requires $\frac{32}{12} \times 0{\cdot}852 + \frac{16}{2} \times 0{\cdot}148 =$ 3·456 lbs. of oxygen per lb. This corresponds to $\frac{3{\cdot}456}{0{\cdot}23} = 15$ lbs. of air per lb. of petrol for complete combustion.

At normal pressure and 60° F. 15 lbs. of air occupy 196 cub. ft.; as the volume of 1 lb. of petrol vapour under similar circumstances is 4 cub. ft., it will be seen that a 15 to 1 mixture, by weight, weighing 16 lbs., occupies 200 cub. ft., 4 of which are petrol vapour. Accordingly this mixture corresponds to 2 per cent of petrol vapour per 100 volumes of charge. In some experiments by Dr. Watson (*Proc. Inst. A.E.*, III, p. 245) it was found that combustion was practically complete when only 14 lbs. of air were used per lb. of petrol, suggesting that the combustion is not as simple as usually assumed. Professors Bone and Hopkinson have observed similar discrepancies. With an air-petrol ratio of 15 to 1 there is, in practice, usually about 1¾ per cent. of free oxygen, and no CO, in the exhaust, a most desirable condition.

Watson's experiments showed also that the maximum thermal efficiency was attained at an air-petrol ratio of about 17½, which is rather too weak a mixture for general use. For these reasons in the above calculation a ratio of 15 to 1 has been assumed.

One lb. of carbon burnt to CO_2 evolves about 14,544 B.Th.U.; if burnt to CO, only 4312 B.Th.U. appear; hence again the importance of getting an exhaust free from CO. One lb. of hydrogen burnt to H_2O evolves about 61,500 B.Th.U., higher value, or 52,830 B.Th.U. lower value.

Hence petrol, having the weight-composition 0·852 carbon and

0·148 hydrogen, has a theoretical maxi of 14,544 × 0·852 + 52,830 × 0·148 = free C and H. The heat value obtaine in a Bunsen burner within a Boys calori to be roundly 18,600 B.Th.U. per lb., B.Th.U. per *pint* at 60° F.

Petrol is very expansible; the ı between 40° F. and 75° F. being about ‹ this range, if *s.g.* denote the specific ‹ sufficient accuracy write:

$$s.g. = 0{\cdot}72\ \{1 - 0{\cdot}0$$

thus, for this spirit, the specific gravit as follows:

Temperature:	40° F.
Sp. gr.:	0·73

This property is sometimes of practic carburettors, as occasionally necessitatin of float between summer and winter use

In Clerk's paper on the 'Principl mined by Exhaust Gas Analysis' (*Pro* it is pointed out that there is in all i best mixture giving for each individu the maximum power concurrently with that it is the function of the carburetto the quantities of air and petrol that th maintained, however the engine may be

In March 1907 the Royal Automob the exhaust gases emitted from the e under some of the conditions of ordin was present in all the cases; the exh tained less than 2 per cent., the remai percentage.

Clerk's tests were made on a 4 Siddeley engine, from which eleven sa taken on three different dates. The tab of the analyses.

Now taking, for example, Pratt's spi of air initially at 60° F. and atmosphe jointly, a volume of 200 cub. ft., be b combustion be reduced to this initial t see from fig. 354 that the final volume will cub. ft. roundly.

TABLE I

EXPERIMENTS BY CLERK ON EXHAUST GASES FROM 18 HP SIDDELEY CAR

Exhaust Gas Analysis by Ballantyne

—	(a) Car standing; engine running as slowly as possible. No load			(b) Car standing; engine running at about 600 revs. per min. No load			(c) Car running on level about 18 miles per hour. Throttle less than half open			(d) Car climbing a hill; engine running about 1000 revs. per min. Throttle full open or over three-quarters open		
	Apr. 23	May 7	July 3	Apr. 23	May 7	July 3	Apr. 23	May 7	July 3	Apr. 23	May 7	July 3
Carbonic oxide, CO	* 0·5	* 0·4	—	* none	3·6	1·8	6·9	4·2	2·4	3·6	3·6	2·2
Hydrogen, H	0·2	0·1	—	none	1·2	0·6	2·4	1·4	0·8	1·2	1·3	0·8
Methane, CH_4	none	none	—	none	0·3	0·2	0·9	0·5	0·3	0·4	0·3	0·3
Hydrocarbon vapours	trace	trace	—	trace	trace	trace	trace	trace	trace	trace	trace	trace
Carbonic acid, CO_2	5·0	6·0	—	6·3	10·8	5·6	9·9	11·5	11·0	11·7	12·2	11·8
Oxygen, O	11·8	10·2	—	11·2	2·2	10·6	0·3	none	2·2	0·1	none	0·6
Nitrogen, N	82·1	83·3	—	82·5	81·9	81·2	79·6	82·4	83·3	83·0	82·6	84·3
Totals	100·0	100·0	—	100·0	100·0	100·0	100·0	100·0	100·0	100·0	100·0	100·0

* Water hot.

Note.—The percentage numbers refer to analysis by volume.

If the petrol has burned completel

exhaust gas consists of 0·852 lb. carbon

CO_2, and 0·148 lb. hydrogen burnt
This has absorbed 3·456 lbs. of oxyg

atmospheric nitrogen amounts to

$11{\cdot}55 + 3{\cdot}12 + 1{\cdot}33 = 16$ lbs.

The heaviness of CO_2 is to that of ai

H_2O, regarded as a perfect gas, is $\frac{}{14\cdot}$

$\frac{14}{14{\cdot}42} = 0{\cdot}971$.

Hence, at atmospheric pressure and
$\frac{3{\cdot}12 \times 13{\cdot}07}{1{\cdot}526} = 26{\cdot}72$ cub. ft.

Similarly, regarded as a perfect
$\frac{1{\cdot}33 \times 13{\cdot}07}{0{\cdot}624} = 27{\cdot}86$ cub. ft.

And 11·55 lbs. of N occupy $\frac{11{\cdot}55}{0{\cdot}9}$
$26{\cdot}72 + 27{\cdot}86 + 155{\cdot}47 = 210{\cdot}0$ cub. ft
ment with the volume as deduced from

Thus when combustion is complete
volume of exhaust should contain about
H_2O (gas), and 74 per cent. of nitrogen.

Usually, however, measurements of
212° F., so that the water vapour is conde
we then have CO_2 and N only, in the pr
85·4 per cent. respectively.

Now 1 cub. ft. of CO_2 at atmospher
$\frac{1{\cdot}526}{13{\cdot}07} = 0{\cdot}11675$ lb., and contains $\frac{3}{11} \times 0{\cdot}$
evolving therefore in its formation 0·03

Similarly, 1 cub. ft. of CO weighs 0·
of carbon, and evolves in burning to CO_2,

And 1 cub. ft. of H weighs 0·0053 lb
60° F., and evolves in burning to H_2O, 0·
(higher value).

Lastly, 1 cub. ft. of CH_4 at atmosphe
0·04245 lb., contains 0·03184 lb. of carbon

and evolves roundly, in burning to CO_2 and H_2O, 1000 B.Th.U. (*v.* Vol. I, p. 117).

Now suppose that on analysis 100 cub. ft. of exhaust gas, measured at atmospheric pressure and 60° F., is found to consist of:

a cubic feet of CO
b ,, ,, ,, H
c ,, ,, ,, CH_4
d ,, ,, ,, CO_2

together with nitrogen, free oxygen, and water.

Then, per 100 cub. ft. of exhaust as above, the heat lost due to the combustible gases discharged is:

$$(326a + 326b + 1000c) \text{ B.Th.U.}$$

The total carbon in the volume under consideration, as determined from the CO, CH_4, and CO_2, is expressed by $0{\cdot}03184\ (a + c + d)$ lbs. If the petrol used have the composition by weight $H = \kappa C$, then there must be in this exhaust a mass of hydrogen existing as H_2O, H, and CH_4, expressed by $0{\cdot}03184\ \kappa\ (a + c + d)$ lbs.; thus the total heat due to the C and H appearing in the exhaust, assumed as initially free and as burnt to CO_2 and H_2O respectively, is expressed by:

$$14{,}544 \times 0{\cdot}03184\,(a + c + d) + 61{,}524 \times 0{\cdot}03184 \times \kappa \times (a + c + d)$$

that is, by $(463 + 1959\kappa)\ (a + c + d)$ B.Th.U.

Hence the fraction of the whole heat of combustion of the fuel regarded as initially free carbon and hydrogen which is lost due to inflammable gases remaining in the exhaust is:

$$\frac{326a + 326b + 1000c}{(463 + 1959\kappa)\ (a + c + d)} \qquad (5)$$

Now for Pratt's spirit, $\kappa = \dfrac{0{\cdot}148}{0{\cdot}852} = 0{\cdot}1737$; the expression (5) accordingly reduces to:

$$\frac{326a + 326b + 1000c}{803\ (a + c + d)} \qquad (5')$$

for this petrol. As an example, consider the case of a very rich mixture of only 9 lbs. of air to 1 lb. of petrol, which gave an exhaust of the composition $a = 12$, $b = 4{\cdot}3$, $c = 1{\cdot}4$, $d = 6{\cdot}7$. In this case the fraction (5′) has the value 0·413, showing that 41·3 per cent. of the total heat of combustion has been wasted in the CO, H, and CH_4 of the exhaust.

In determining the expression (5), 100 cub. ft. of exhaust at atmospheric pressure and 60° F. has been assumed, to fix the ideas. But

it is obvious that the value of
other pressure and temperatu
represent the volume percenta

The denominator of (5) is
fuel burns as free carbon and h
which case, e.g. for Pratt's spi
of 20,210 B.Th.U. (lower) is o
chemically combined in the
furnish a value of only 18,600
would appear that the coeffic

$$\frac{18,600}{20,210} \times 803 = 740.$$

Dr. Watson (*Proc. Inst. A.*
the results of a number of exhau

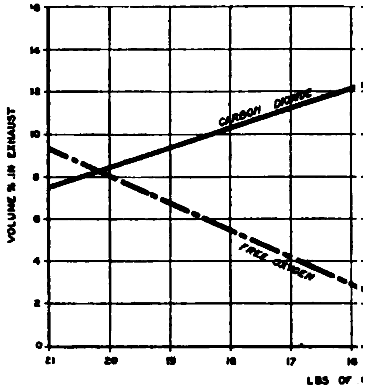

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fig. 355, which well resumes his
except in a very small region wh
from about 13¾ to 14¼, free O
exhaust; if there is free O, ther

He expresses the opinion tha
of air to petrol to be inferred,
practical purposes, from an anal
percentages of O, CO, and CO_2.

With carbonic oxide are as
(CH_4) in the constant proportio

Per cent. of free H =
Per cent. of CH_4 =

From these proportions and the

Table II of exhaust gas composition has been obtained for different ratios of air to petrol; the two columns to the right show the percentage of the total heat of the fuel carried away in the combustible gases of the exhaust, and evolved in the engine respectively, as given by the expression (5′):

TABLE II. (*From Watson*)

Ratio of $\frac{\text{air}}{\text{petrol}}$	*a* CO	*b* H	*c* CH_4	*d* CO_2	*e* O	*f* N by diff.	Total heat of fuel Per cent. lost in exhaust	 Per cent. appearing in engine	 *a*
9	12·0	4·3	1·4	6·7	0·0	75·6	41·3	58·7	44·7
10	9·6	3·5	1·2	8·2	0·0	77·5	35·6	64·4	38·5
11	7·3	2·6	0·9	9·6	0·0	79·6	28·7	71·3	31·1
12	5·0	1·8	0·6	11·1	0·0	81·5	20·8	79·2	22·5
13	2·6	0·9	0·3	12·5	0·0	83·7	11·5	88·5	12·4
14	0·4	0·1	0·0	14·0	0·4	85·1	1·4	98·6	1·5
15	0·0	0·0	0·0	13·0	1·7	85·3	0·0	100·0	0·0

The extreme right-hand column headed '*a*' shows the percentage of heat thrown away in the exhaust calculated by taking the coefficient in the denominator of (5′) as 740 instead of 803. It must be remembered also, that in obtaining all the above figures the water vapour present in the exhaust has first been condensed.

Ballantyne's results as to the constancy of the ratio of the free H and of the CH_4 to the CO enable expressions (5) and (5′), *ante*, to be much simplified. For we may write $b = 0{\cdot}36a$, and $c = 0{\cdot}12a$; whence, on reduction (5) becomes:

$$\frac{563a}{(463 + 1959 \cdot \kappa)\,(1{\cdot}12 \,.\, a + d)} \tag{6}$$

while (5′) takes the simple form:

$$\frac{0{\cdot}7}{1{\cdot}12 + \dfrac{d}{a}} \tag{6′}$$

involving only the ratio of the CO_2 to the CO in the exhaust. From (6′) the proportion of the total heat of the fuel wasted in the combustible gases of the exhaust can be quickly estimated.

Many attempts have been made to estimate the proportions of air to petrol in the mixture supplied to the engine by the carburettor from an examination of the exhaust gas analysis; so far, however, such analyses have not been sufficiently complete to enable this to be

For the same petrol this ratio should be constant, and equal to K, the ratio by weight of H to C as determined by a direct analysis; see Table IV (*infra*).

Again, the air used bears to the joint weight of the C and H the ratio:

$$\frac{14{\cdot}42 \times \frac{100}{79} f}{b + 2c + 0{\cdot}532f - 2\left(\tfrac{1}{2}a + d + e\right) + 6\left(a + c + d\right)}$$

which becomes on reduction:

$$\frac{\text{Air}}{\text{Petrol}} = \frac{18{\cdot}25f}{0{\cdot}532f + 6{\cdot}32a + 4d - 2e} \qquad (8)$$

TABLE IV

Exhaust analysis, per cent. volumes				Air/Petrol		Value of K	
a	*d*	*e*	*f*	By direct measurement	From Eq. (8)	From analysis of petrol	From formula (7′)
CO	CO_2	O	N				
12·0	6·7	0·0	75·6	9	9·66	0·1737	0·1823
9·6	8·2	0·0	77·5	10	10·5	0·1737	0·1849
7·3	9·6	0·0	79·6	11	11·46	0·1737	0·1896
5·0	11·1	0·0	81·5	12	12·46	0·1737	0·1910
2·6	12·5	0·0	83·7	13	13·76	0·1737	0·2000
0·4	14·0	0·4	85·1	14	15·2	0·1737	0·1880
0·0	13·0	1·7	85·3	15	17·2	0·1737	0·2050

In Table IV results from (7′) and (8) are exhibited for the cases given in Table II, and it will be noted that the calculated values are roughly approximate for the richer mixtures, but become considerably too high for ordinary working proportions. This is due to the usual method of analysis not accounting for all the carbon existing in the exhaust.

Hence it does not appear practicable to infer the proportions of the mixture supplied by the carburettor to the cylinders from analysis of the exhaust as usually conducted.

Dr. Watson has pointed out, however, that if the ratio of air to petrol in the mixture be estimated from the hydrogen only, as deduced from the results of an ordinary analysis, and if K be independently determined for the fuel (petrol) used, then values are obtained by calculation which correspond better with the results of direct measurement, particularly for very rich or very poor mixtures.

ratios, a result arising from the approximate character of the usual assumptions made as to the constitution of the exhaust.

It may be noted that over the range from 11 to 19 the dotted straight line in fig. 356 well resumes the results of calculation from Eq. (9′); the true (i.e. measured) values being represented by the full line at 45° to the axes, we have for these experiments, and within this range, the practical result:

$$\text{Corrected ratio} = 1{\cdot}06 \times \text{Ratio calculated from Eq. (9′)}$$

i.e.

$$\text{Corrected } \frac{\text{air}}{\text{petrol}} \text{ ratio} = \frac{2{\cdot}86f}{0{\cdot}532f - 0{\cdot}4a - 2\,(d + e)} \qquad (9'')$$

The experiments of Prof. B. Hopkinson and Mr. Morse (*Proc. Inst.*

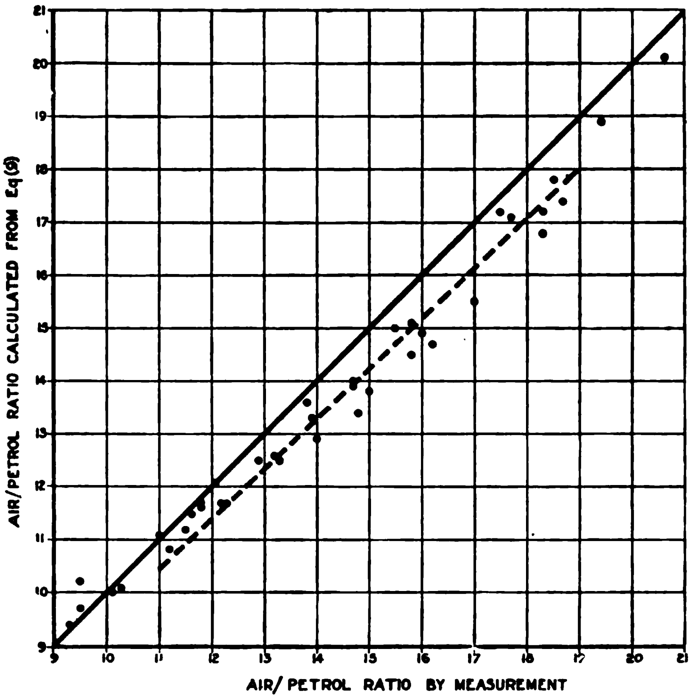

FIG. 356

A.E., III, p. 284) showed that maximum thermal efficiency and maximum power appeared to occur practically together at an air-petrol ratio of about 14, at full load both at 720 and at 1220 revolutions per minute.

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pipes through which air was drawn b
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spirit.

Butler endeavoured to secure unif
vessel containing a diametral diaphra
extending downwards to within ½ in. of
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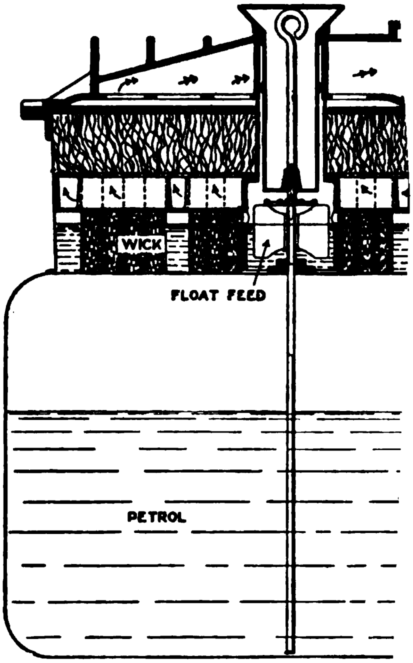


FIG. 357

their upper ends being supported by c
the wicks reached to the bottom of
effectually screened its surface from th
gradually deteriorated during working,
which had to be drained off.

The only surface carburettor survi
the present time (1911) is that of the
an illustration is given in fig. 357.

The large lower vessel is the petrol t
tank divided into three compartments

Of these, the lowest, having a capacity of about 1 gallon, contains petrol maintained at a constant level by the light metal float and valve indicated; the petrol was originally raised into this compartment by a hand pump, and later by exhaust pressure supplied through the pipe marked 'pressure inlet'; latterly a small pump with oscillating cylinder of the type familiar in model steam engines has been employed. In this petrol, and reaching to the bottom of the compartment, are a number of wicks which extend upwards through partitions in the second chamber into the topmost chamber, which is thus loosely filled; the upper ends of the wicks terminate against a gauze screen at the top of the uppermost chamber, the floor of which is perforated. Hot air is drawn by the engine suction into the middle chamber, and passing through these perforations, traverses the upper loose portion of the wicks, which are charged with petrol by the capillary action; a richly carburetted air accordingly passes through the gauze screen and enters the mixing chamber at the top of the carburettor; this is diluted to the necessary degree by fresh air admitted through the cylindrical mixing valve shown in the upper right-hand corner of the diagram, and the product thence passes to the motor through the pipe indicated; the mixing valve diminishes the vapour and increases the fresh air, or *vice versa*, simultaneously. In Messrs. Lanchester's hands this carburettor has proved very successful; the main objection to it would appear to be its bulk and weight; for their standard 38 HP engine, for example, the weight is about 30 lbs.

It will be noted that the free surface of the petrol is sealed from the action of the air, and it is claimed that, owing to this and the general disposition of the carburettor, all 'fractionation' of the petrol is avoided; it is obvious also that there is no possibility of any unevaporated petrol reaching the cylinder, i.e. that this is a 'gas' as distinguished from a 'spray' carburettor; also that the action is not affected by the presence of particles of grit or water in the petrol.

Excepting in the Lanchester engine, the float feed spray carburettor with induced jet action is now in universal use in all petrol engines from the smallest bicycle motor to the largest racing machine, the standard arrangement still closely resembling the original Maybach model of 1893, more fully described later on in this chapter.

Reference may first be made, however, to a very simple method of carburetting air which has been more largely used in America than in England and consists in its most elementary form in connecting the petrol supply with a perforation in the inlet valve seating. When the valve opens the petrol issues from this perforation in a fine spray, and mingling with the inrushing air, supplies the necessary explosive charge; the rate of flow of the petrol was usually regulated by a

screw-down needle valve capable o
valve ' of this type was fitted to th
engine.

This method is, however, of Britis
in 1881 for his two-cycle engine, gas
in the valve seat. It was also use
their heavy oil engines before 1895.

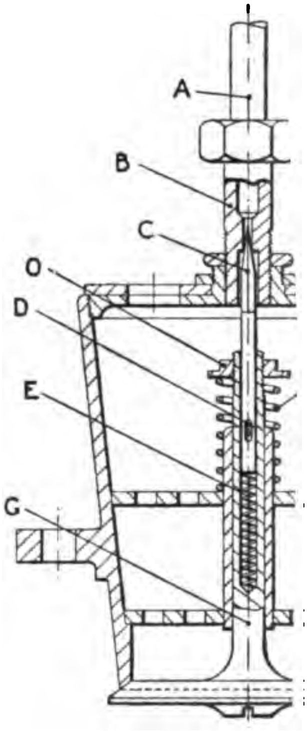


FIG. 358

The admission of liquid petrol di
tend to incomplete vaporisation, and i
more easily controlled mixture variou
form appeared. Thus in the Carlton n
supply through A is controlled by a nee
stem of the inlet valve G. When the e
is held up to its seating by the fine
valve stem ; D is a vertical slot in t
a pin passes connected to the inlet v

spring. On the suction stroke of the piston the inlet valve G opens against the spring L and descends through a short distance without opening the needle valve; when, however, the pin bottoms in the slot D the needle valve is opened, and petrol then issues into the valve chamber. This petrol is taken up by the air entering through the regulator grating M, and becomes completely mixed and vaporised by passing through the perforated diaphragms K and H, before entering the cylinder. The petrol supply can be adjusted by means of the screwed nozzle B; the air supply is controlled by the movable grating M; and the inlet valve spring compression by the flanged nut O. An adequate air suction during the open period of the needle valve is ensured by the pin and slot D, the arrangement causing the inlet valve to open earlier and close later than the needle.

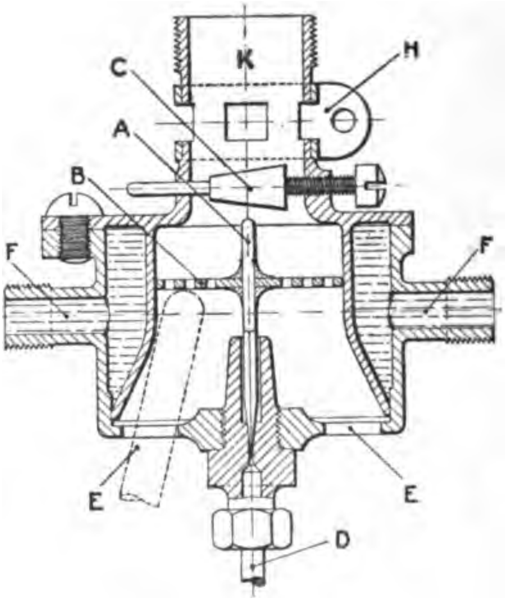


FIG. 359

In the Capel, Creek, or Johnston mixing valve the petrol was supplied to a perforation in the seat of an automatic, spring-controlled, flat-faced air valve, the mixture being then passed through a spiral chamber having an exhaust-heated core, thus becoming completely vaporised before passing through the inlet valve to the cylinder.

An extremely simple mixing valve that has been successfully employed in small engines (e.g. the de Dion bicycle motor) is illustrated in fig. 359. The petrol supply D is controlled by a needle valve, A, carrying a light metal perforated disc, the. weight of the needle valve and disc On the suction stroke air enters throu the valve chamber, passes through the K ; when a sufficient velocity is attained are raised, and petrol then issues into the air through B and thus completely valve is limited and regulated by the c able sleeve, H, admits and regulates extra tion is assisted by exhaust-jacketing connecting branches. To obtain a rich m the air supply at H is cut off and the ne through one of the holes E as indicated simple and inexpensive, but it is difficult t of petrol with it.

In the single-cylinder Cadillac engine ment is adopted, a gravity valve with dis of air during suction. The disc is of ver of small vanes are pressed out of its surf these causes the disc and needle valve even wear at the seat and prevents leaka

Among other carburettors of this gen the Alderson, Blake, Crossley-Hulley, Knapen, Lepape, Lucas, New, Oldsmo Union. These have all given fairly s been largely used, especially in America line ' engines ; they are in general relat to manufacture ; they are, however, action, nor capable of the necessary ni them to adequately fulfil the exacting speed petrol car engine.

Another mode of carburation that attention was that of supplying a definit for each working charge. This method who used a pump with mechanically o much difficulty was experienced in pre liquid fuel and air ; the packing of t much trouble.

Butler designed an improved appa piston and pump plunger so arranged th into the carburetting chamber in propo

The well-known Gobron-Brillié ca further attempts in the same direction equidistant cells on its outer surface casing containing an annular channel by which the cells were filled with the liquid fuel. The plug was actuated by a ratchet and pawl in such manner that the contents of a cell became exposed to the inrush of air at each suction stroke of the engine, thus causing carburation to occur.

As already stated, however, all these earlier devices have now yielded place in the high-speed petrol engines of automobiles to the constant-level float-feed induced jet type of carburettor in one or other of its very numerous forms.

Of these the earliest is the 'Inspirator,' invented and patented by Butler in 1889; a diagram in section is given in fig. 360.

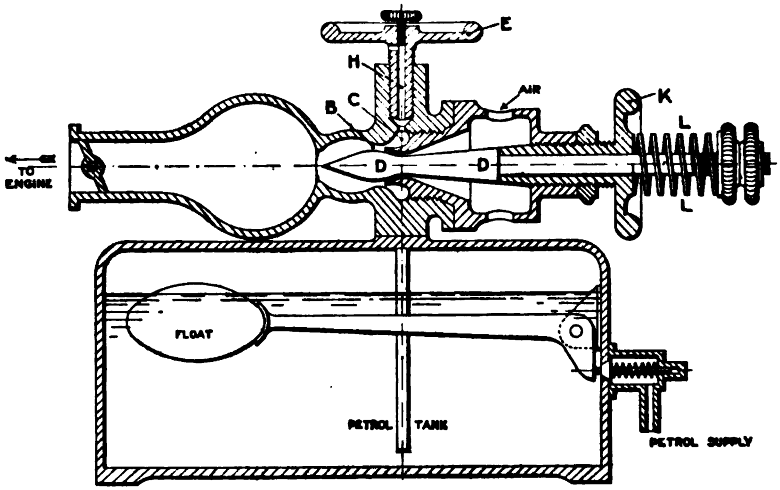


FIG. 360

The level in the petrol tank is maintained constant by a ball valve of the usual cistern type; a pipe connects this tank with the annular space C, into which the petrol is drawn by the suction caused by the inrush of air through the vena contracta B during the charging stroke of the engine. The petrol reaches the annular space C by way of the screw regulator valve E, and is atomised in its passage through this valve by the central air jet H. The central rod or 'air-plug' D D can be adjusted in position by the screw K, and is also controlled by the spring L L so as to automatically vary the area of the vena contracta at different engine speeds and thus maintain an approximately constant suction.

Maybach's famous carburettor of 1893, though held by the courts to be anticipated by Butler's invention of four years earlier, is the protofunnel could be regulated by means of a perforated diaphragm indicated diagrammatically in fig. 361. The top of the needle valve projects through the float chamber cover, and is protected by a screwed cap; in the arrangement shown, the buoyancy of the float is utilised to forcibly close the needle valve, while when the float sinks the weighted outer ends of the small bell-cranked levers E cause the needle to rise.

Maybach considered also the heating of the carburettor to assist vaporisation when using heavy hydrocarbons, and proposed admitting additional air, at will, to the mixture after leaving the carburettor, in order to regulate or stop the engine. He thought that with his design of carburettor a mixture of constant proportion would be obtained at all engine speeds; he reasoned that the velocity of the air and consequently the suction in the choke tube would vary in proportion with the engine speed, and hence the proportion of air to petrol would remain constant. This, however, is not the case; the phenomena of flow for gases and liquids differ, and the general result in this case is that at low speeds the mixture is extremely weak, while at high speeds it becomes too rich.

Fig. 362 is an instructive diagram, showing the results of some experiments by Dr. Watson, which exhibits the nature of the action very clearly; the curve O C A shows the relation between the quantity of air actually passed through the carburettor and the suction in inches of water; the curve D C B indicates the quantity of air which, mixed with the petrol actually found to issue from the spray nozzle, would produce a mixture of constant composition.

It will be noted that in these experiments, until the suction exceeded half an inch of water no petrol at all issued from the nozzle, and that the mixture was weak from half an inch to one and a half inches of suction; beyond this the mixture is too rich. These results were obtained with constant suction; in an actual engine the action is further complicated by the rapid fluctuations in the suction, inertia effects both of the air and petrol being then introduced.

The experiments of Bickford on the efflux of kerosene from spray nozzles, and of Prof. Morgan on that of petrol, show the velocity to vary as the square root of the pressure difference between the reservoir (float chamber) and nozzle outlet, and Wimperis shows ('The Internal Combustion Engine,' pp. 266 *et seq.*) that, with certain simplifying assumptions, the velocity of the air through the choke tube may be similarly expressed. Wimperis shows also that the velocity of efflux of the petrol, v, is related to that of the air through the choke tube, v_0, agreeably with the equation:

$$v^2 = 0{\cdot}0017\, v_0^2 - 2gh \qquad (10)$$

where 0·0017 expresses the ratio of the density of air to petrol, and h

Owing to the tendency of the ordinary induced spray carburettor to give too weak a mixture at low speeds and too rich a mixture at high speeds, a common practice is to so adjust the apparatus that the mixture is somewhat rich at low speeds to facilitate starting and enable the engine to be run slowly; the increased richness at higher speeds is then prevented by admitting extra fresh air at some point between the carburettor and the throttle valve of the engine. This extra air is sometimes hand-controlled, but in most normal touring cars it is admitted automatically through a spring loaded valve which is so set as to open when the suction in the inlet pipe increases; in this way the mixture delivered to the engine is rendered more uniform in composition.

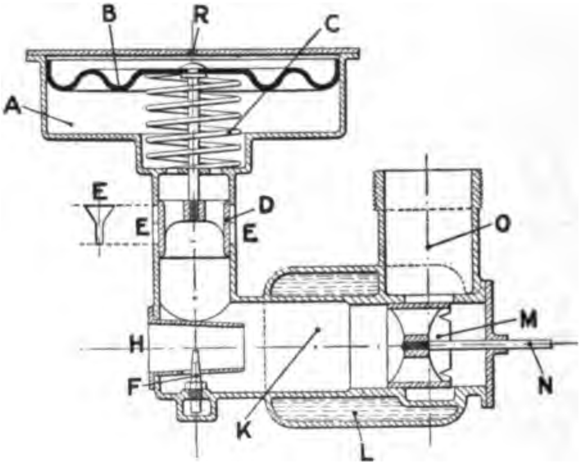


FIG. 363

The well-known Krebs carburettor (fig. 363) was one of the first embodying a carefully designed automatic extra air supply; the device consisted essentially of a closed drum, A, containing a flexible metallic diaphragm, B, supported by a helical spring, C; the centre of this diaphragm is connected by a light rod with the cylindrical extra air valve D; F is the petrol nozzle; the main air supply enters at H, traverses the nozzle, taking up the petrol spray, and enters the hot-jacketed mixing chamber K; thence it passes the cylindrical throttle M and reaches the engine by way of the inlet branch O. The throttle M is provided with V-shaped depressions in its working edge, so that when the throttle is closed a minimum amount of mixture necessary to keep the engine running slowly may still be permitted to pass. The cover of the drum contains a very small hole, R, by which the upper

surface of the diaphragm is kept in c
sphere ; as the engine suction increas
atmospheric pressure depresses the di
extra air is then admitted to the mixing
shaped holes E ; the hole R in the dru
to cause a 'dash-pot' action and pre
diaphragm.

In the Grouville and d'Arquembourg
annular chamber surrounds the mixing
the bottom of this annular chamber con
diameter each normally covered by a
vary in diameter. As the speed of the
suction, varies, more or fewer of these b
and the supply of extra air to the mixing

In the Renault carburettor the ext
weighted disc valve whose movemen
working in a small dash-pot filled with

In the La Buire design a sliding pla
moved transversely by hand. The slot
a cylindrical valve moving vertically be
the extra air valve. Motion of the pla
valve and admits extra air to the mix
other direction shuts off the extra air.
latter direction lifts an additional valve
directly into the inlet pipe close to the c
downhill the engine is cooled, and oil is n
into the combustion chambers, through
when running.

In Smith's 'Perfecta' carburettor th
immediately above the spray nozzle, an
which passed into the spray orifice and
engine speed the whole air supply passe
rounding the nozzle ; in normal working
proportion to the suction, and the petrol
with it through the corresponding move

Another design in which the petrol fl
of the Thornycroft 'T. T.' carburettor,
in fig. 364. On the suction stroke of th
throttle B being open, warmed air en
tube A A, taking up a charge of petrol sp
and passes through the triangular orifice
to the engine ; extra air is admitted in inc
inlet shown as the throttle is more widel
the issue of petrol from the nozzle is inc

piece F attached to the throttle lifting the needle valve in the spray nozzle as shown. This carburettor is not jacketed, but the main air supply through A is very effectively warmed before admission; the level of the petrol in the spray nozzle is maintained by a float-feed box of the usual type, not shown in the illustration.

In the Unic carburettor the spring-controlled automatic extra air valve is connected with a small piston working in a cylinder filled with glycerine, to prevent 'dancing.'

In the Austin carburettor the extra air is admitted through an automatic spring-loaded valve to a space surrounding the horizontal mixing tube, the walls of which are perforated; the whole carburettor in this case is very simple in form.

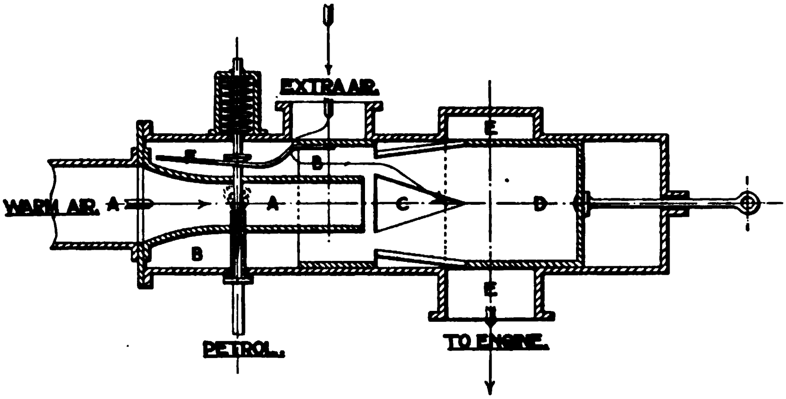


FIG. 364

The devices for admitting and regulating the extra air supply to the mixture are extremely numerous, but in recent practice the admission is effected in a large number of cases through a simple light automatic spring-controlled 'mushroom' valve with ebonite or leather seat, placed between the carburettor and the inlet valve of the engine.

Some designs, e.g. the Rover No. 1 and the Sthenos, seek to attain uniformity of mixture by causing the area surrounding the spray nozzle to automatically vary with the suction. For example, in the well-known 'Sthenos' carburettor, shown diagrammatically in fig. 365, A is the air inlet and B the spray nozzle; C is a fixed perforated dished diaphragm, while D is a movable expanded choke tube furnished with a skirt which, when in its lowest position, is seated upon the dished diaphragm C. At low engine speeds all the air passes through the circular orifice K K, the choke tube being then in its lowest position; the considerable contraction in the area of the air stream in passing KK induces sufficient suction for a proper working mixture light; E is an adjustable by-pass for the mixture when the throttle is closed. As the throttle is opened air is enabled to pass in increasing degree outside the tube C. The carburettor is adjusted in manufacture by varying the diameter of the small holes in this tube, and also by the size of the gaps cut in the throttle, so that the mixture is kept as constant as possible at all degrees of opening; three petrol

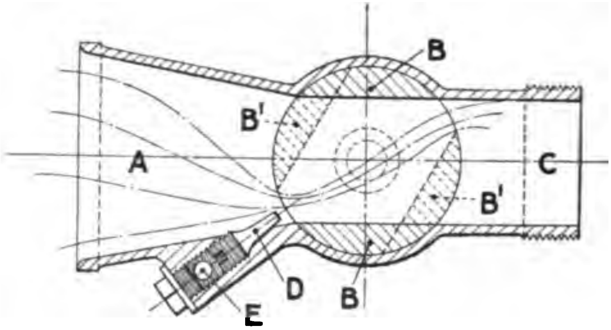


Fig. 366

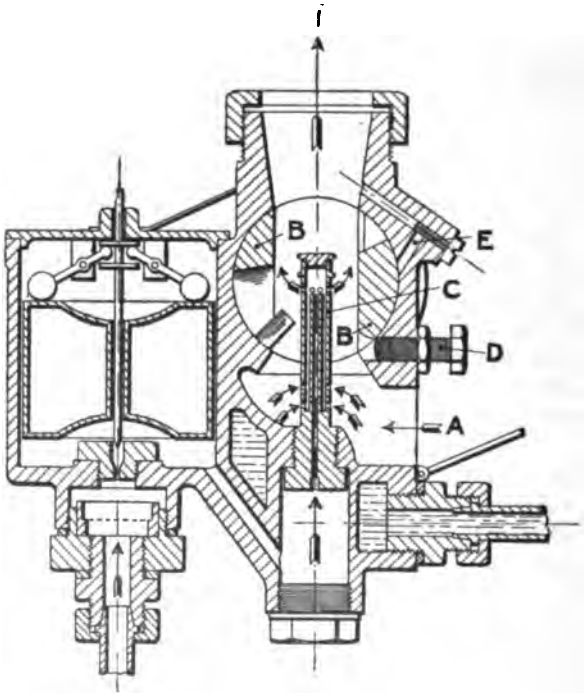


Fig. 367

nozzles differing in diameter by 0·002
carburettor, the most suitable size fo
mined by actual trial. A hinged flap
temporarily cut off so as to 'flood'
the engine. It will be noted that th
of the carburettor is jacketed in thi
ferably by hot water, but if thermo-sy
mended that the exhaust gas be used
jacketing are that it is irregular in act
rupted by oily carbonaceous deposit
and chamber.

Considerable success has also be
wherein the petrol and air supplies
controlled. Thus in the 'Amac' car c
with a number of small orifices and
cylindrical throttle valve that on this
of orifices and the area of the air strean
together. In the Hillman-Coatalen de
tained two groups of three small ho
operated by a lever enabling the petrol s
In the Chenard-Walcker carburettor a
finely tapered needle which entered the
the effective area of the orifice; at incre
thus admitting more air and simultane
accompanying withdrawal of the ta
orifice. In the Scott-Robinson design
into the spray nozzle was carried by
as this drum rose with increased sucti
and the petrol increased with that of t
again, the extra air valve carried a tap
nozzle orifice and was so arranged that
was proportionately withdrawn, causing
to increase together.

The above are but a few of many f
ments that have been employed; one of
of the positively controlled modern car
by Messrs. White & Poppe, Ltd. In th
but the area of efflux is varied with the a
drical throttle, so that suitable proporti
at all times. The nozzle is considerabl
and is drilled through the top with an e
nozzle is fitted with a sleeve containing
moves with the throttle C C in such mann
the petrol orifice is completely uncov

direction of the ingoing stream of air. The throttle works within a cylindrical sleeve, F F, so fixed in the initial adjustment of the carburettor as to determine the required degree of richness of the mixture.

A section of the complete apparatus is given in fig. 369; it will be seen that the throttle is operated through a vertical stem carrying a lever, and that the sleeve surrounding the spray nozzle is directly connected to it; tightness between the sleeve and nozzle is obtained by means of a helical spring compressed between the top of the throttle and cover of the spray chamber; this cover is formed in one piece with the mixture regulating sleeve F F. The level of the petrol in the nozzle is maintained constant by a float feed of the usual type. The carburettor is not jacketed, and all the air enters through the inlet D (fig. 368); but special stress is laid upon the effective warming of the air, and it is recommended that the intake be from the heated surface of the exhaust pipe in the manner illustrated in fig. 370. The sleeve A is about 15 ins. long, and open at both ends; at B cold air can be admitted at will for regulating the temperature of the supply in warm weather; the total area of the holes in B, as also the cross-sectional area of the air pipe itself, is made twice that of the carburettor air port to prevent any wire-drawing and consequent over-enrichment of the mixture; with the same object in view the air pipe is kept as short and free from bends as possible.

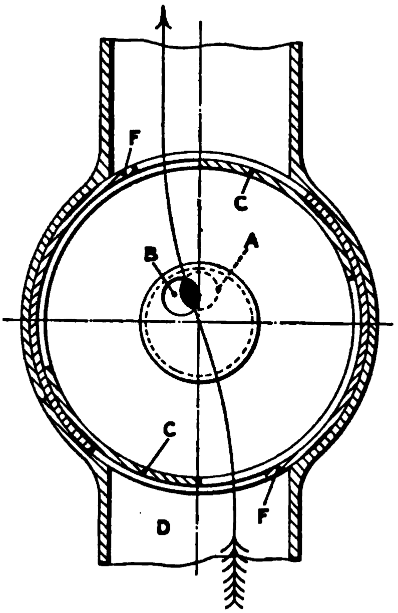


FIG. 368

The White & Poppe carburettor is the outcome of much experience and experimentation, and good results are obtained with it; reference may here be made to the performance of the 15·9 White & Poppe engine, and of the 20 HP Vauxhall engine described in Chap. VIII.

The authors are indebted to Mr. P. A. Poppe for the following test results obtained in June 1911 from a 30 mm. standard White & Poppe carburettor; the exhaust analyses were made by Mr. Stacey Jones, B.Sc., of Coventry, with an Orsat apparatus. The engine was a four-cylinder, 3·54 ins. × 4·33 ins. White & Poppe; in the first series of tests the speed was maintained constant at 1360 revolutions per minute, the corresponding piston speed being about 980 feet per minute, and the maximum brake horse-power developed 29; the results are exhibited in fig. 371, and in the accompanying table; it will be noted that, except at very light loads, carbon monoxide was practically absent:

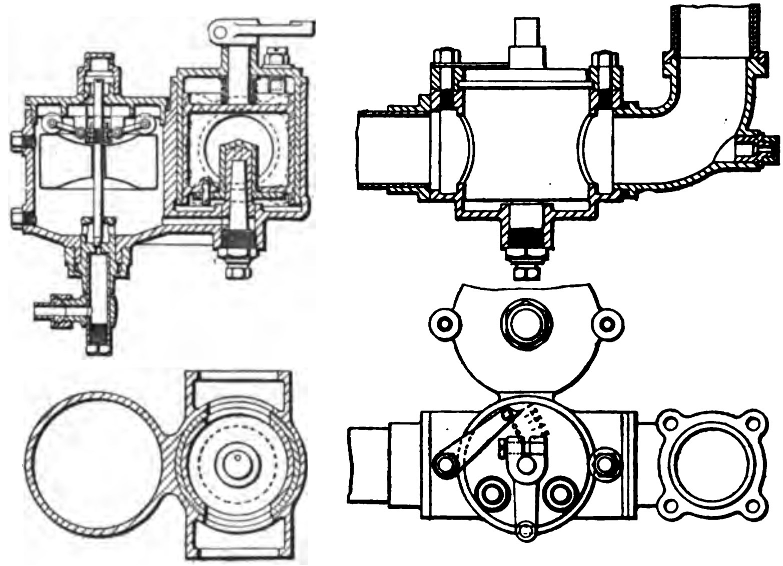


FIG. 369

TEST OF A WHITE & POPPE 'NO. 30' CARBURETTOR. CONSTANT ENGINE SPEED OF 1360 R.P.M.

Brake HP	Pints of petrol		Volume per cent. in exhaust of		
	per hour.	per BHP hour	CO_2	O	CO
0	5·3	∞	10·9	0·9	1·8
5	6·8	1·36	13·0	0·8	0·3
10	8·9	0·89	13·3	0·65	0·1
15	11·25	0·75	13·3	0·55	trace
20	13·6	0·68	13·3	0·4	trace
25	15·9	0·636	13·3	0·3	—
29	17·8	0·615	13·3	0·2	—

In a second series of tests with the same engine and carburettor the revolution speed was varied, the full power at each speed being obtained; the results are exhibited in fig. 372 and in the accompanying table. In this case also combustion appears to have been complete throughout, only a trace of carbon monoxide being detected at any time, while the free oxygen varied from 0·8 per cent. to 0·2 per cent. These results show that, when properly adjusted, the White & Poppe carburettor is capable of giving practically perfect combustion at varying powers and speeds.

Test of a White & Poppe 'No. 30' Carburettor at Various Engine Speeds

Revs. per min.	Brake HP	Pints of petrol		Volume percentage of		
		per hour	per BHP hour.	CO_2	O	CO
550	10·4	6·8	0·654	13·2	0·8	0·2
700	14·2	9·0	0·633	13·2	0·7	0·2
900	18·4	11·4	0·620	13·2	0·55	0·2
1100	22·7	14·0	0·617	13·2	0·42	0·2
1400	29·0	17·8	0·614	13·2	0·2	0·2

In another class of carburettor, originally introduced by M. Leon Bollée, uniformity of mixture is aimed at by using more than one spraying nozzle, a common arrangement in the earlier designs being to provide one nozzle with a constricted choke tube giving a rich mixture for starting or slow running, and a second giving the normal proportions at ordinary speeds; in some cases the starting nozzle was cut out when the second came into operation; in others both were used together.

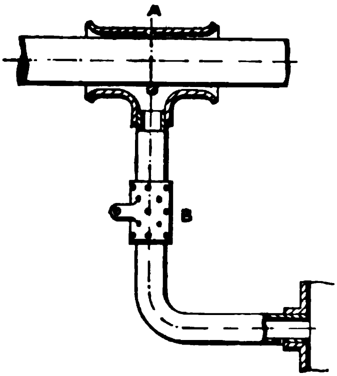


Fig. 370

Thus, in the Zenith carburettor, illustrated in fig. 373, the inner main nozzle A is surrounded by a second annular nozzle, B B, and there is a further small orifice just opposite the edge of the disc throttle valve, as shown at H. This and the annular nozzle are in permanent communication with the small chamber C, open to the air, through the passages G and E respectively; the chamber C is supplied with petrol

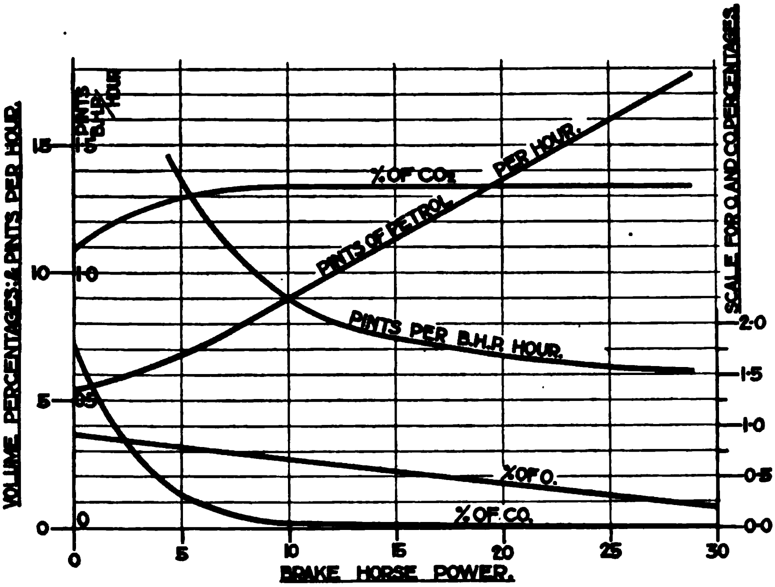


FIG. 371

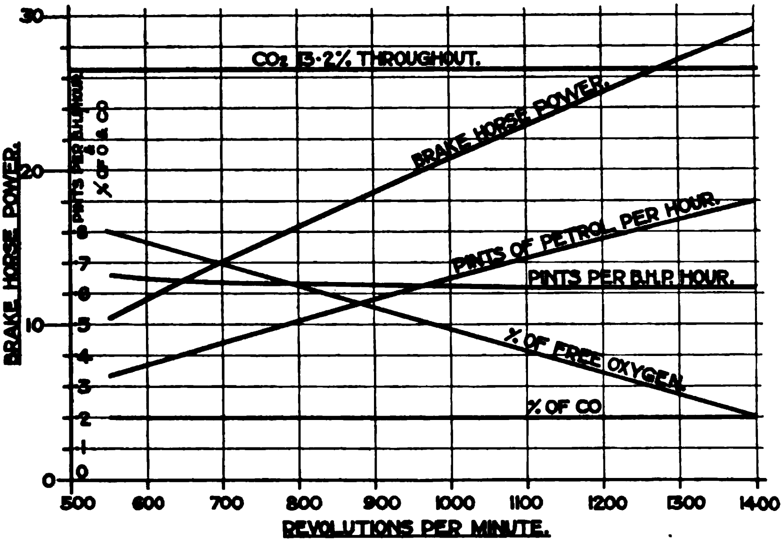


FIG. 372

at a slow but constant rate from the float chamber through the narrow duct D.

The arrangement facilitates starting and slow running at light load; at starting the chamber C is full of petrol, and accordingly a rich and readily ignitible mixture is obtained, as the main and annular nozzles and the orifice at H all then supply spray to the entering air.

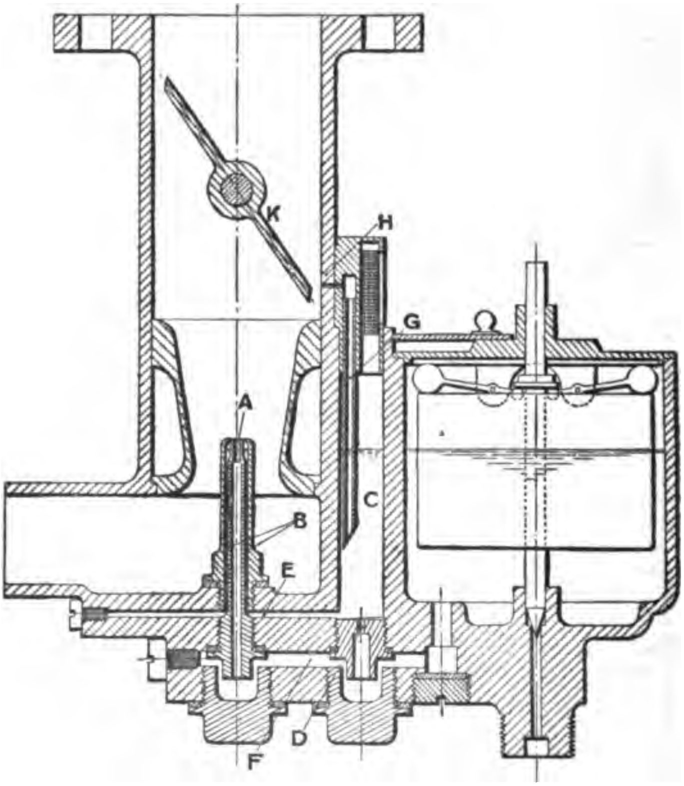


FIG. 373

For slow running at light load the throttle is nearly closed, and the orifice H alone then furnishes the necessary spray for carburetting the induced air.

At normal speed and load the chamber C becomes depleted, and the carburation of the air is mainly dependent upon the inner main nozzle A only.

In the Mors a secondary small nozzle, having its own inlet, communicates with the main induction pipe between the throttle valve and the engine; this secondary nozzle keeps the engine running slowly when the throttle is shut. These are not, however, strictly carburettors of the multiple jet type proper, the auxiliary jet being adopted for starting and slow running only.

In the Roydale carburettor also there were two unequal nozzles, each having its own hot-jacketed choke tube, the smaller alone being used at slow speeds; as the throttle opened the second jet was gradually brought into action, and in normal running the two nozzles worked jointly. Similarly, in the Rolls-Royce there are two nozzles with independent air supplies delivering into a common mixing chamber; the choke tube of one of these is covered by a disc valve; as speed and suction increase this valve is automatically raised, thus bringing the second jet gradually into action and maintaining proper proportions of mixture. And in the Gregoire there are also two nozzles having similar functions.

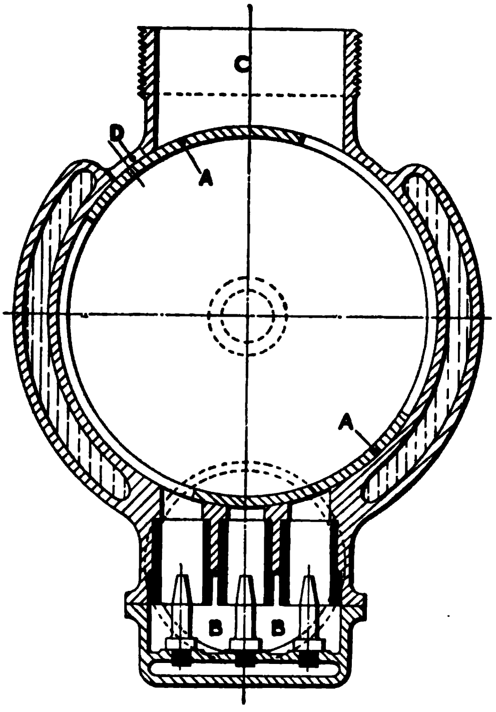


FIG. 374

Of three-jet carburettors that known as the H.P. is simple in design, and a diagrammatic sectional view is given in fig. 374; the cylindrical throttle A A on being moved in a counter-clockwise direction successively uncovers the petrol jets; the air inlet is at B; the jets are all alike, each with its own choke tube. The throttle mixing chamber is hot-jacketed, and the branch C communicates with the engine; on turning the throttle sufficiently clockwise the supply of gas is cut off, while a further rotation in the same direction uncovers a hole, D, communicating with the atmosphere, through which cold air is enabled to pass directly into the engine when the car is running downhill, thus not only cooling the cylinders, but avoiding carbon deposit, smoky exhaust, and fouled plugs due to oil being drawn past the pistons through the engine pumping against a vacuum.

In the Trier and Martin carburettor three nozzles are also employed, the first of which is the largest, and is used at low speeds; these are placed in line and deliver into a cylindrical chamber in which is a sliding valve connected with a sliding throttle of larger diameter; as the latter is opened, admitting more air, the jets are successively uncovered. The mixing chamber is hot-jacketed.

The Wolseley Company also use a three-jet carburettor; the first jet is very small and used only for very slow running; a further throttle opening cuts this out and brings the second jet into operation; this is of a size to give economical running at ordinary speeds. At full throttle the third jet comes into action together with the second; this third jet is relatively large, and increases the richness of the mixture, power being regarded as the primary requisite at full throttle, high efficiency being then a secondary consideration; there is also a hand-controlled extra air valve.

Nothing better emphasises the extreme difficulty of the problem of perfect carburation, especially for the engines of automobiles, than the immense number and great diversity of the devices that have appeared with this object during the past twenty years. One of the latest and admittedly most successful of the multiple jet type that has been produced is the invention of Mr. A. G. Ionides, and is appropriately named the 'Polyrhöe' (many jet) carburettor; in this the principle is carried to an extreme limit, a very large number of exceedingly fine jets being employed.

In the Polyrhöe carburettor the intake of air is through a horizontal rectangular opening variable both as to its length and breadth; the length automatically varies with the engine suction, while the breadth is hand adjustable, as explained later.

Along one edge of the rectangular intake there is a continuous line of very minute jet orifices having each a cross-section of about one ten-thousandth of a square inch. The essential feature of the invention is shown in the accompanying diagrams, figs. 375 and 376. A A is the row of fine jet orifices ranged along one edge of the rect-

angular opening ; B is a mova
air intake is varied in accordan
fig. 375 the length of the intake
has increased to C′ D′, which i

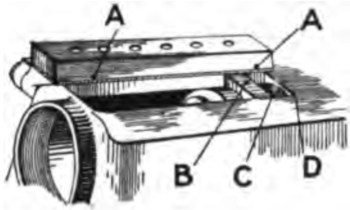


FIG. 375

is delivered only from those jet o
of air rushes, whence it is clear th
therefore the petrol supply is pro
the rectangular intake, that is, t
proportions of the mixture are
The minute jet orifices are rectan
slots cut in the edges of brass pl
thickness ; one of these ‘jet pl
scale in fig. 377 ; several such pla
the slots alternating, to form the
jet orifices along the edge of th
angular intake of the carburett
378 illustrates to a much enlarge
one arrangement adopted. Th
under the action of the suction
by the inrush of air past them, iss
the minute orifices in a horizonta
tion across the intake aperture ; t
at once taken up by the air stream
on to the engine.

It will be noted that cold air is
at the rectangular intake ; the bo
hot-jacketed, and it is important
effective in order that complete v
place as far as possible within the
best performance ; the jacketing n
cooling water, or exhaust gas may

The mode by which constanc
attained at all degrees of opening

and it remains next to be seen how the degree of richness may be varied over the whole range of working. This is effected by the use of a regulator slide, E E (fig. 379), by which the *width* of the rectangular air intake may be varied at will; reduction of the width increases the suction along the line of jets exposed to the air stream, and thus increases the richness of the mixture by a constant amount for all degrees of air opening. This regulator slide is hand operated through a Bowden wire connection with an index and dial fixed on the dash-board; the quality of the mixture is thus immediately adjustable at all times while the engine is running, and variations necessitated by differences in petrol density, air temperature, humidity, &c., &c., can be promptly made.

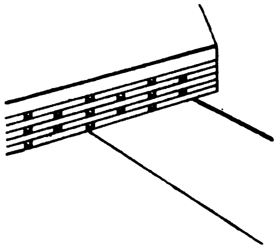


FIG. 378

A noteworthy point in the design is that, owing to the column of petrol communicating with each jet orifice being very small, it has very little mass, and due to this and the relatively low velocity of efflux of the petrol, inertia effects are practically eliminated, the jets responding instantly without perceptible lag to rapid changes in the air supply.

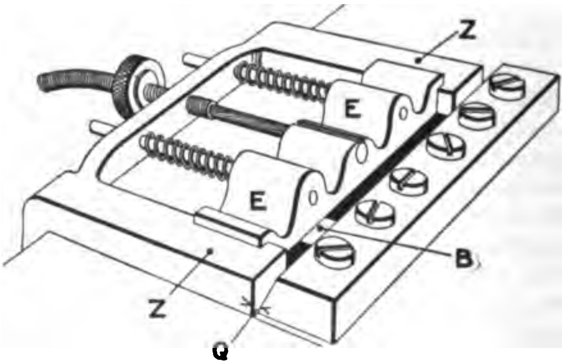


FIG. 379

The complete carburettor is shown in longitudinal and transverse sections in figs. 380 and 381; A is the line of jet orifices, O the rectangular intake; the air enters at O, inducing a discharge of petrol from the jets traversed, enters the mixing chamber N, hot-jacketed as shown at S, and passes thence into the inlet branch R, and so to the engine; the quantity of mixture admitted to the branch R is regulated by the hand-contolled throttle M. H is a hollow trunk connecting the block B with a piston, K, working easily but with air tightness in the cylinder X. A light helical spring, L, within this cylinder causes the piston to move to its extreme right-hand position and thus close the air intake, when the engine is not running; on starting, a partial vacuum is created in the mixing chamber N; as the outer end of the cylinder X is in communication with the mixing chamber through the hollow trunk H, while the inner end is constantly open to the atmosphere, the partial vacuum in N extends also through H into the space L, and the excess atmospheric pressure on the inner face of the piston K accordingly causes this, with the attached trunk H and block B, to move towards the left, thus opening the air intake O until a balance is attained between the vacuum and the spring L. W is a simple non-return disc affixed to the inner end of the hollow trunk H; it allows air to be readily withdrawn from the cylinder into the mixing chamber, thus facilitating rapid opening of the air intake, but it offers some resistance to the ingress of air, and thus causes a slower closing of the air port and exercises also a steadying or 'dash-pot' action upon the movements of the system B, K, H.

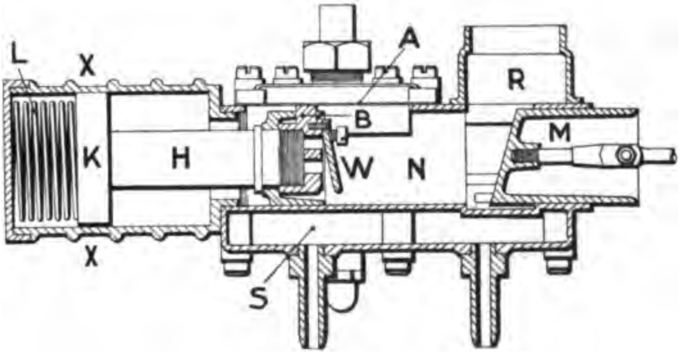


FIG. 380

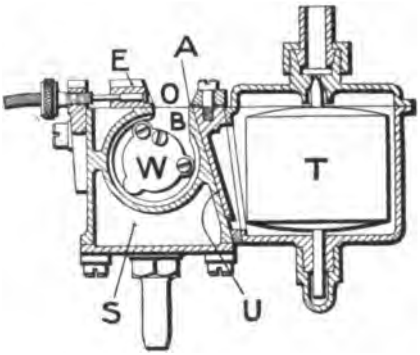


FIG. 381

The float feed T is of very simple design; the petrol supply enters at the top and the needle is directly connected to the float; a screwed cap at the bottom of the chamber enables the petrol to be drained off when required, and also gives access to the needle valve for turning this on its seat to remove grit, &c. U is the line of petrol ducts from the float chamber to the horizontal line of jet orifices A; provided that the level of petrol in the float chamber is high enough to well cover the inlet to these ducts near the bottom, and not so high as to cause flooding from the jet orifices, its actual intermediate position is not a matter of much importance.

Fig. 382 shows diagrammatically the petrol ducts U in more detail, and also indicates the large fine-meshed gauze diaphragm Y through which the petrol must pass before reaching the ducts U, thus preventing any of the small jet orifices from becoming obstructed by particles of grit.

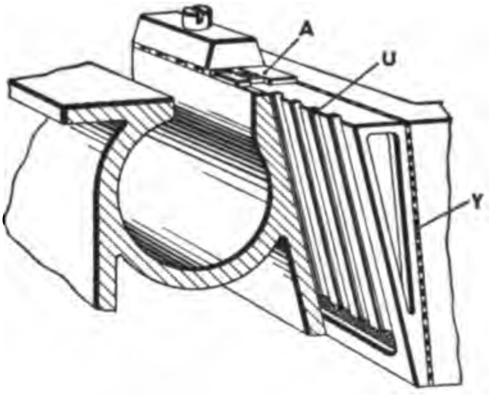


FIG. 382

One last point remains to be mentioned, viz. the nature of the adjustment for 'no load,' so that the engine will run slowly and smoothly when light.

The regulator slide E E (fig. 379) is borne in guides Z Z, which reach nearly but not quite, across the whole width of the air intake; a small gap Q is left between the end of Z and the face of the line of jet orifices, and the whole guide piece Z Z may be moved in a direction parallel to the line of jet orifices so that the number of these which continue in operation when the automatic block B has closed the main air supply can be varied at will; the number of operative jets is diminished until the mixture supplied is as weak as possible consistently with regular slow running of the engine; the guide piece Z Z is then locked in the position thus determined by a pair of set screws passing through slots in this guide into the back of the carburettor. This one easily made preliminary adjustment includes the necessary corrections for the type of ignition, location of plugs, degree of compression, extent of air leakage along inlet valve stem guides, &c., for each individual engine; the adjustment should be made when the engine and carburettor are fully warmed up.

The simple means by which the richness of the mixture is controlled from the dashboard, through the regulator slide E E, is a most favourable feature; the adjustment for maximum power is obtained with ease and certainty while the engine is running; for normal running it is recommended that a mixture rather weaker than that giving maximum power be employed; this—as already pointed out earlier in this chapter—results in a somewhat improved efficiency and produces also an innocuous exhaust.

It may be mentioned that the throttle valve M (fig. 380) can, after shutting off the mixture, be moved still further towards the left so as to open communication direct between the inlet branch R and the atmosphere, allowing cold air to be drawn into the engine when the car is running downhill.

During the summer of 1910 some experiments were carried out by the *Automotor Journal* with a Polyrhöe carburettor fitted to a four-cylinder, 4 ins. × 4½ ins., 30 HP Cadillac car[1] engine; a complete account of these tests and the results obtained is given in the same journal for August 13, 1910.

The engine had previously been used with several carburettors of different types, so that it was possible for the experimenter to compare the relative performances with the Polyrhöe. It was found that, without altering the hand-controlled adjustment for richness, the engine pulled fully as well as the best previously attained under the most favourable conditions, and that in respect of slow running and rapid increase of engine speed considerably better results were obtained than was usually the case; when the throttle was suddenly opened an instant change to full power was experienced.

In the matter of fuel economy, about 37 ton-miles per gallon were obtained on ordinary dry English country roads when maintaining 'a good average pace'; this is a very satisfactory figure. On the Brooklands track at 30 miles per hour the fuel consumption on a run of eleven miles with three passengers up was found to be at the rate of 26½ miles per gallon; this leaves little to be desired on the point of economy.

The tests further showed that very effective warming of the mixing chamber is essential in order to maintain a characteristically good performance.

Ten samples of exhaust gas were collected, all at the same setting of the richness adjustment, and under varying conditions of running at Brooklands; the analyses were made by Mr. H. Ballantyne, and his results are given in the accompanying Table VII:

---

[1] Weight of car, without passengers, was from 3000 to 3200 lbs. in running order.

TABLE VII

TABLE OF EXHAUST GAS ANALYSES AT VARIOUS LOADS AND SPEEDS, AND ALSO AT THE MOMENTS OF SUDDEN ACCELERATION

	Test No.	1	2	3	4	5	6	7	8	9	10
	Speed of engine (r.p.m.)	300	300	600	950	300	550	950	1250	1580	500
	Approximate equivalent car-speed (m.p.h.)	—	10	20	30	10	17	30	40	48	16 and up
	Degree of load	none	very light	light	light	full	full	full	full	full	accelerating
Analysis figures per cent.	Carbon monoxide (CO)	0·8	0·2	0·8	nil	nil	0·2	0·2	0·2	nil	nil
	Hydrogen (H)	0·3	0·1	0·3	nil	nil	0·1	0·1	0·1	nil	nil
	Methane (CH_4)	0·1	nil	0·1	nil	nil	nil	nil	nil	nil	nil
	Carbon dioxide (CO_2)	10·1	13·2	13·4	13·5	13·4	13·8	13·5	13·4	13·8	6·8
	Oxygen (O)	5·8	1·4	0·2	1·0	1·2	0·2	0·2	1·2	0·6	10·9
	Nitrogen (N)	82·9	85·1	85·2	85·5	85·4	85·7	86·0	85·1	85·6	82·3
		100·0	100·0	100·0	100·0	100·0	100·0	100·0	100·0	100·0	100·0

It will be seen that samples were taken at engine speeds ranging from 300 to about 1600 revolutions per minute, and from light load at the lower speed limit to full load at the upper; in the last test the sample was collected at moments of suddenly opening the throttle from closed to full open position, in order to ascertain the extent to which the mixture was thereby affected.

Applying Eq. (6′) to these results, we obtain the following figures as the percentages of the total heat of the fuel wasted in the exhaust:

Test No.:	1	2	3	4	5	6	7	8	9	10
Per cent. of waste:	5	1	4	0	0	1	1	1	0	0

Excluding Nos. 1 and 10, where the conditions were not those of normal running, it will be seen that, excepting the somewhat anomalous result in test 3, the combustion was practically complete within the engine over a range of speed from 600 to 1600 revolutions per minute and from light to full load. The exceedingly small proportion of CO is a most noteworthy feature, the average of the whole ten tests being but 0·24 per cent., which is only 8 per cent. of the quantity evidenced in the early R.A.C. experiments with twelve different car engines as already referred to.

In respect of rational design, excellence of performance, and ease and accuracy of adjustment at all times to meet any variation in running conditions the Polyrhöe expanding carburettor takes a high position.

CHAPTER X

HEAVY OIL ENGINES (INCLUDING THE DIESEL ENGINE)

HAVING in Chap. VI discussed briefly the chemical and physical properties of the petroleum oils, the reader is in a position to consider the mechanical arrangements of oil engines. The lighter oils being so easily vaporised were naturally first used in the early forms of oil engine. With oils of a specific gravity less than 0·76 and a flashing point as a rule lower than the ordinary atmospheric temperature 60° F., such as benzine, naphtha, gasolene, or 'petrol,' the problem of producing an inflammable mixture capable of being drawn into an engine cylinder, compressed, and exploded, is so simple that no complicated considerations trouble the inventor in producing engines when run at low speeds. The earlier oil engines accordingly used such light oils.

Early Oil Engines.—The earliest proposal to use oil as a means of producing motive power by explosion appears to be that of Street, whose English patent was taken out in the year 1791. The first practical petroleum engine, however, was that of Julius Hock, of Vienna, who produced an engine in 1870. This engine operated on the old non-compression system, and took in a charge of air and light petroleum spray during part of the forward stroke of a piston, ignited that charge at atmospheric pressure by means of a flame jet, and so produced a low-pressure explosion similar to that of the Lenoir gas engine. In 1873 Brayton, an American engineer, produced an oil engine (*v.* Vol. I, p. 21). In that engine heavy oil, it is true, was used having a density sometimes as high as 0·85, but this oil was crude unrefined oil flashing at about atmospheric temperature. The engine was not a practical success, but it was the first compression engine using oil fuel instead of gas.

Shortly after 1876, when the Otto engine came into use, several engines of that type appeared operated by air gas obtained from 'gasolene' or 'petrol' by drawing air through or over the volatile spirit, thus charging it with vapour and forming an inflammable mixture. The air thus 'carburetted' was drawn into the

cylinder with a further supply of
mixture, and compressed and ignite
cycle gas engine.

Spiel Engine.—The Spiel petrole
of the Otto cycle practically used whi
vaporising apparatus. In this en
was injected directly into the cylinc
mixing with the inrushing air becam
on compression an explosive mixtu
precisely as the gas mixture in th
never became popular, mainly becau
that time regarded as dangerous ; t
affecting the transport and storage of
large scale practically impossible.
small petrol engines, particularly in
this simple device of a ' mixing valv
charge (see Chap. IX) ; in general t
the ' float feed ' carburettor, but w
cost and simplicity are prime consider
still followed.

ENGINES USING KEROSENE

Petroleum oils with a flash point exc
treatment in order to obtain an explo
required varies to some extent with
the oil used. Many engines are now
and Russian kerosenes, crude and re
and shale oils ; these range in specific
low degree of volatility of these pet
vaporising devices to produce the
with air. Such engines are now exte
cultural, and marine service, but a
propulsion of automobile vehicles,
varying demand for power in this class
impossible to obtain regular and clean

With petrol the cylinder receives a
and heavier oils the necessary prelimi
re-condensation necessitate the supply
to the cylinder ; the volumetric effici
order to prevent pre-ignition lower
necessary in cases where air alone is
such cases, the thermal efficiency of t
the calorific value of kerosene is usually

petrol, engines designed to use either fuel at will in general develop only about 85 per cent. as much power from kerosene as from petrol when the kerosene is heated in a 'vaporiser' before entering the engine cylinder.

Classification of Heavy Oil Engines.—Engines using heavy oils as fuel have been classified by Mr. E. Butler according to the mode in which the explosive mixture of oil vapour and air is obtained, substantially as follows:

Class A.—Vaporisers employed with the normal form of internal combustion engine cylinder, including:

Type a_1.—Engines with separate exhaust-heated vaporisers wherein a 'mist' of oil and air is sprayed by an air-pump as in the Priestman, Griffin, and Halliday engines. Or in which the piston suction is utilised to induce the mixture of oil spray and air into the heated vaporiser as adopted by Thornycroft, Ivel, Barker, Butler, and Constantinescu.

Type a_2.—Engines with separate lamp-heated vaporisers, as in the Howard kerosene engines with special forms of oil force pump, or the Crossley and Dudbridge designs in which a measuring cup is filled and emptied at each induction stroke. In the Roots-Vosper, again, the oil gravitates to a chamber fitted with a sliding rod carrying a measuring cavity by which the correct charge of oil is admitted to the mixing chamber at every induction stroke; the mixed oil and air pass through a cellular hollow-walled lamp-chimney vaporiser before entering the cylinder. Other engines falling in this category are the Globe and the Hellier.

It is important in this type that the quantity of oil injected into the vaporiser should be carefully regulated to the needs of the engine; the advantage of lamp-heated vaporisers is that the engine may be run very regularly and economically at light loads, similarly to an ordinary gas engine; the disadvantage is that, however carefully designed the lamp or lamps may be, they invariably demand frequent careful cleansing and attention; accordingly the tendency in oil engine design is towards the employment of one or other of the several forms of lampless ignition.

Type a_3.—Engines fitted with a separate exhaust-heated paraffin carburettor. Few paraffin carburettors have survived; a typical design is that known as the Moorwood-Bennett, of which a description and illustration appear later in this chapter.

Class B.—Vapourisers necessitating a special form of explosion chamber to the engine cylinders; this class includes:

Type b_1.—Engines having a combined vaporiser and explosion chamber, forming part of the cylinder head. The Hornsby-Akroyd engine is the best example of this type; the oil fuel

is injected directly into a 'hot-bu
at a dull red heat by the success
forms an extension of the cylinde
municating therewith by a narow n
and well, and once started is indepen
heating. Other engines of this type
Crossley (crude oil), Norris, Mietz & \
general, an oil force-feed injection pı
fuel direct into a vaporiser forming p
In the earlier Norris designs the oil wa
iser surrounded by and open to t
chamber.

Type b_2.—Engines having a vaporis
head, and either explosion, exhaust, (
type are frequent and increasing in nuı
the Ruston, National, Tangye, Field
Capitaine, &c. Many of these makers i
lamp heating of the vaporiser and ign
of cases at the present time the lamp
only. Where there are prolonged peri
sometimes necessary, however, to resoı
maintain the temperature of the vapoı

Type b_3.—Engines depending for v
entirely upon the very high temperature
compression pressure ; illustrated in th

Type a_1: *Priestman Oil Engine.*—
first engine capable of using safe
heavy burning oils (normal kerosenes) ; it was built as a single-cylinder engine in sizes from 1 to
11 nominal horse-power, and with
two cylinders up to 25 horse-power.
Although this engine could be
worked with heavy Scotch 'paraffin' of sp. gr. 0·820 and flash
point about 150° F., the best
results were obtained with 0·8
kerosenes flashing at about 100°
F. The oil spraying principle
adopted was that so well known
and widely used by perfumers.
Fig. 383 shows such a spray producer
an air blast issuing from the small no:
end of a tube B, thus creating within B a

FIG

in the reservoir C, accordingly rises in B, and overflowing from the top is met by the jet from A and at once converted into a very fine spray or mist by its action. If such an apparatus be filled with a petrolum oil as Royal Daylight, or Russolene, the oil will also be blown into fine spray, and this spray may be ignited by a flame, and will burn, when suitably adjusted, with an intense blue, non-luminous flame. The earlier inventors often expressed the idea that an explosive mixture could be prepared without any vaporisation whatever by simply producing an atmosphere containing inflammable liquid in extremely small particles distributed throughout the air in such proportion as to allow of complete combustion. The familiar explosive combustion of lycopodium, and the disastrous explosions caused in the exhausting rooms of flour mills by the presence of finely divided flour in the air, have also suggested to inventors the idea of producing explosions for power purposes from combustible solids. Although, doubtless, explosions could be produced in that way, yet in oil engines the production of spray is only a preliminary to the vaporisation of the oil. If a sample of oil be sprayed in the manner just described and injected in a hot chamber also filled with hot air, then the oil so sprayed will at once pass into a state of vapour within that chamber, although the air may be at a temperature far below the boiling-point of the oil. The spray producer, in fact, furnishes a ready means of saturating any volume of air with heavy petroleum oil to the full extent possible from the vapour tension of the oil at that particular temperature. The oil engines about to be described are in reality explosion gas engines of the ordinary Otto type with special arrangements to enable them to vaporise the oil to be used. In the Priestman engine oil was forced by means of air pressure from an oil reservoir along a pipe connected to B (fig. 384); issuing from the fine jet orifice *a*, it was met by compressed air admitted into an annular chamber *b* surrounding the jet orifice, and injected in the

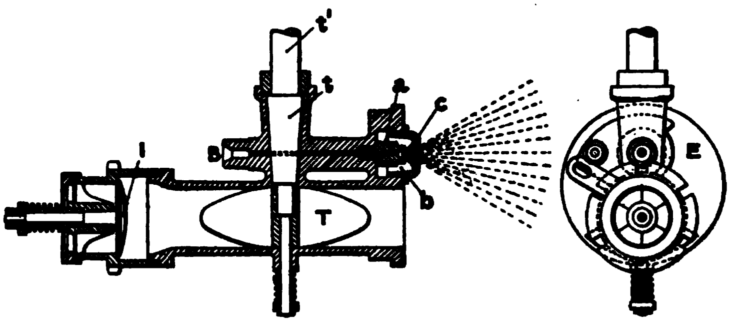


FIG. 384

form of a very fine spray or ' mist '
an exhaust-heated vaporiser E, wher
on the suction stroke of the piston th
through the automatic valve I, was
pressed, and fired electrically.

To start the engine a hand-pump w
pressure to force the oil through the
The vaporiser was heated by a blow-lan
sufficiently high to vaporise the spray n
turned by hand and the engine starte
then be extinguished, the heat of the
the exhaust jacket. Governing was
and oil supply ; the throttle valve T a
(fig. 384) were carried on a common spin
gear. The air and oil supplies were t
it was sought to maintain a constant pr
loads there was a working stroke every o
of the throttling of the charge on red
very light loads fell so low that the e
of the non-compression type, resulting
rise in the fuel consumption per IHP un

Tests and Oil Consumption.—Profes
Priestman engine at the Royal Agricultu
The engine tested was a 4½ HP nomin
12 ins. stroke, normal speed 180 revolut
was Broxburn Lighthouse, a Scotch p
destructive distillation of shale. Its d
point about 152° F. Analysis of the oil

Carbon . . . . . .
Hydrogen . . . . . .
Deficiency . . . . . .

By calculation the heating value is 1
the total heat evolved, including heat of
liquid state. The principal results given

Indicated HP . . . . .
Brake HP . . . . .
Duration of trial . . . .
Mean speed (revolutions per minute)
Mean available pressure (lbs. per sq. in.)
Explosions per minute . . .
Oil consumed per IHP per hour (lbs.)
Oil consumed per BHP per hour (lbs.)

The heat account is :

Total heat shown by indicator	12·67
Heat given to jacket water	53·39
Exhaust waste and other losses	33·96
	100·02

In 1892 Professor Unwin made another trial of a 5 HP Priestman oil engine at Hull, in the course of which he used both Russolene oil and Daylight oil. The engine was of the same dimensions as the Plymouth engine, that is 8·5 ins. cylinder and 12 ins. stroke. The volume swept by the piston per stroke was 0·395 cub. ft., and the clearance space in the cylinder at the end of the stroke was 0·210 cub. ft. The small air-compressing pump supplying the spray producer discharged 0·033 cub. ft. per stroke. The total weight of the engine was 36 cwt., including a flywheel of 10 cwt. The principal results obtained were as follows :

	Daylight	Russolene
Indicated HP	9·369	7·408
Brake HP	7·722	6·765
Mean speed (revolutions per min.)	204·33	207·73
Mean available pressure (lbs. per sq. in.)	53·2	41·38
Oil consumed per IHP per hour	0·694 lbs.	0·864 lbs.
Oil consumed per BHP per hour	0·842 „	0·946 „

With Daylight oil the explosion pressure was 151·4 lbs. per sq. in. above atmosphere, and with Russolene 134·3 lbs. The terminal pressure at the moment of opening the exhaust valve with Daylight oil was 35·4 lbs., and with Russolene 33·7 per sq. in. The compression pressure with Daylight oil was 35 lbs., and with Russolene 27·6 lbs. pressure above atmosphere. Analyses were made of the samples of Daylight and petroleum by Mr. C. J. Wilson, F.C.S.

	Daylight Per cent.	Russolene Per cent.
Carbon	84·62	85·88
Hydrogen	14·86	14·07
Oxygen	0·52	0·05
	100·00	100·00
Specific gravity at 60° F.	0·7936	0·8226
Flashing point	77° F.	86° F.

The total heat of combustion of Daylight oil calculates out at 21,490 B.Th.U., and for Russolene at 21,180 B.Th.U.

Professor Unwin calculates the amount of heat accounted for by the indicator as 18·8 per cent. in the case of Daylight oil, and 15·2 in the case of Russolene oil. Fig. 385 is a diagram taken by Professor

Unwin, and published in his paper re
Engineers in 1892. The largest dia
the diagram in dotted lines is half
diagram shows the card given by th
load. The various particulars of clea
pressure of compression, and stroke
the illustration. From these figures i
oil engine worked on a consumption of c
per hour, and 0·842 lb. of Daylight oil

Professor Unwin states that the
was insignificant in quantity, being

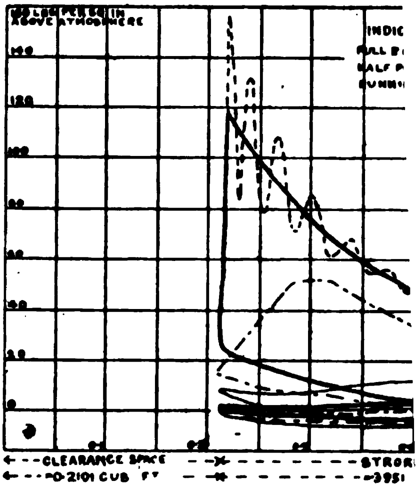


FIG. 385.—Priestman Oil Engi

each of the two trials in which it was m
and battery, primary bichromate cells be
gas engine.

Samuelson Oil Engine.—The Samuel
under the Griffin patents; as in the I
sprayed into an exhaust-heated vapori
by lamp-heated hot tube, and the engi
cutting off the air and oil supply and also c
the speed became too high.

Fig. 386 is a section of the Griffin
enters by way of the passage A, and disc
thereby creating a partial vacuum in
between the air nozzle and the oil no
to the oil-supply chamber B¹ by way of

to a plunger stem C¹. The air pressure, when admitted, forces down the plunger C¹, and thus opens the valve C against the pressure of the spring. Oil then passes up the pipe B² from the chamber B¹, and is discharged with the air from the nozzle A¹ in a state of fine spray.

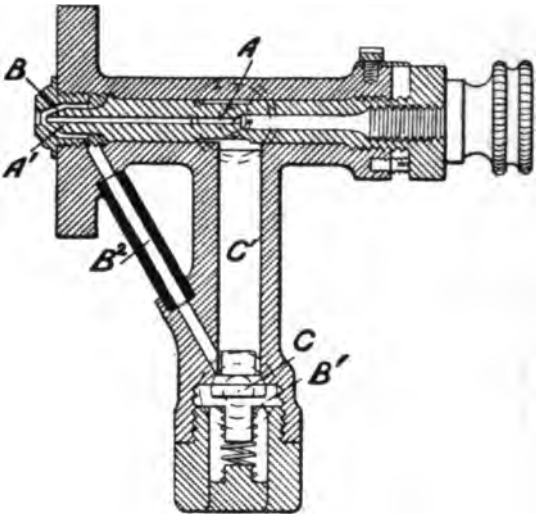


FIG. 386.—Samuelson (Griffin) Oil Sprayer

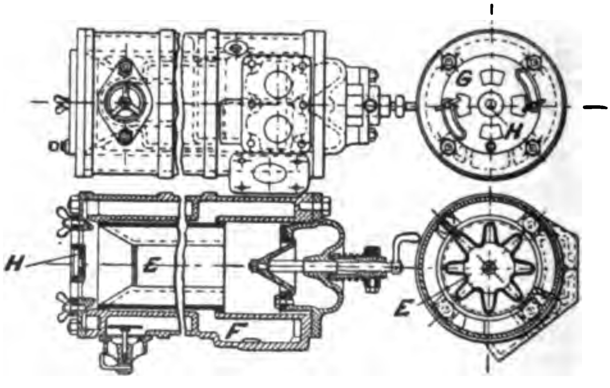


FIG. 387.—Samuelson Engine Vaporiser

Whenever the air pressure is removed from the plunger C¹, the spring forces the valve to its seat, and cuts off the oil supply. The air pressure is maintained at from 12 to 15 lbs. above atmosphere by a pump driven from an eccentric on the valve shaft.

The vaporiser is shown in longituc
and end elevation at fig. 387. E is t
outline, and surrounded by the exhau
to the vaporiser from the atmosphe
plate G, and the spray nozzle is attac
the spray into the centre of the vapc

In the early Priestman and Samue
oil is sprayed into the vaporiser by m
by a pump, but in many recent desi
and the engine suction alone used t
vaporiser, the ordinary float-feed spı
petrol engines being now largely empl

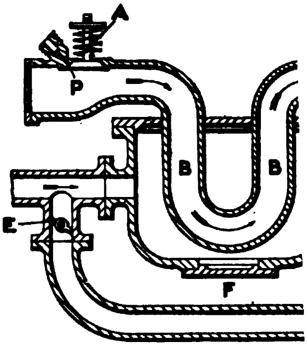


Fig. 388

The Thornycroft Paraffin Vaporise
croft & Co. have devoted much attenti
'paraffin' vaporisers of the induced su
early design of this firm included a spı
fine orifice in its seat connected with tl
suction caused this valve to open an
kerosene spray and air to the vaporis
of a U-shaped tube surrounded by exh
shown diagrammatically in fig. 388;
inlet valve in the seat of which is an orifi
sprays when the valve opens. The r
passes into the exhaust-jacketed U-tu
C is an adjustable regulator by which
to obtain a suitable explosive mixture,
which the speed of the engine is contro

The temperature of the U-tube is r
exhaust pipe containing a throttle valv

closed by an easily removable door, the U-tube is heated by a blow-lamp for starting the engine.

This vaporiser, though satisfactory at full loads, possessed the following defects:

(1) Liability to flood when starting.

(2) Liability to flood when running light.

(3) Want of engine controllability due to the difficulty of adjusting the kerosene supply to the varying requirements of the engine with this type of mixing valve.

A further source of trouble arose when running light through the exhaust being then unable to maintain the U-tube at a sufficiently high temperature to completely vaporise the fuel; liquid kerosene consequently collected in the bottom of the U-tube; on opening the engine throttle this liquid was sucked into the cylinder and was liable to cause stoppage at the engine by fouling the ignition plugs.

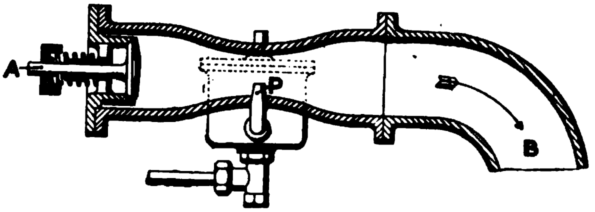


FIG. 389

To overcome the troubles arising from flooding at starting and when running light Messrs. Thornycroft replaced the mixing valve by an ordinary float-feed spraying-nozzle of the petrol type as in the Cottrel vaporiser, but with rather larger spraying orifice for kerosene; the arrangement is shown diagrammatically in fig. 389; A is the spring-supported air valve and P the kerosene nozzle; the suction of the engine, as before, induces a rich mixture of air and spray into the U-tube B. With this device it was easy to start the engine on petrol, and by means of a two-way cock switch over to kerosene as soon as the cylinders and vaporiser became sufficiently heated; by using the exhaust by-pass and thus keeping the vaporiser cool, the engine could be run on petrol continuously if desired, though at some small sacrifice of economy. Or again, if the use of petrol were objected to, the engine could be started by heating the U-tube with a blow-lamp for 15 or 20 minutes as in the earlier arrangement (fig. 388).

The difficulty of running light for any length of time remained, however, in the U-tube design, and in later vaporisers Messrs. Thornycroft have employed an exhaust-heated straight tube containing a broad-bladed helix through which the air-spray mixture passes; the helix causes the mixture to pursue a whirling path and the vaporisation is much more completely effected. No accumulation of liquid kerosene occurs with this type, the whirling motion causing any unevaporated spray to be distributed over the heated inner surface of the vaporiser and at once boiled off, and the engines run satisfactorily at light load and low speed for prolonged periods. A section showing the latest form of this device is given in fig. 390.

A A is the vertical cast-iron, helix-containing, externally-ribbed, exhaust-jacketed tube through which the engine suction induces a current of air which enters at the inlet B and passes along the constricted tube C, containing the float-fed spraying nozzle D, hand-regulated by a needle valve E. It will be noted that there are two float chambers, F_1 and F_2; one contains petrol and the other kerosene; the spray nozzle is served with petrol for starting purposes and until the engine and vaporiser are warmed up; the petrol is then cut off and the kerosene simultaneously turned on by a small hand lever attached, as shown, to the two-way cock K.

When petrol is not used a single float chamber suffices; in this case the vaporising tube A A must be first heated by a blow-lamp for 15 or 20 minutes through the openings G_1 and G_2. The exhaust gases from the engine pass through the branch H and thence around the jacket surrounding A A, making their exit by the branch M; the plate valve L L enables the temperature of the vaporising tube to be regulated; in the position shown in fig. 390 all the exhaust passes through the jacket, but by moving the valve a greater and greater proportion is enabled to pass directly into the outlet branch M; finally, when the valve is in the position indicated by the dotted lines, the exhaust all passes direct to M and the vaporising tube is not heated.

The rich mixture of vaporised kerosene and air passes along the pipe O and receives an addition of fresh, cool air from the belt PP through the automatic spring-supported air valve R; thence by way of the triangular ports SS the explosive mixture passes to the branch T, and so to the cylinders. It will be seen that the sliding regulating valve V simultaneously controls the quantity of mixture passing to the engine and the admission of extra air through the orifices W.

Messrs. Thornycroft favour kerosene as fuel on the score of cheapness and safety, especially for decked boats; a considerable number of fishing-boats are now running with their 'paraffin' engines. Starting can always be effected by blow-lamp heating, but in many cases, e.g. as in fishing-boats, it is important to be able to get under way very quickly, and accordingly in such cases it is usual to fit a tank on deck containing two to four gallons of petrol, on which the engine is started and run for from five to ten minutes in order to get everything well warmed up; the kerosene is then turned on. In general, Russian oil is preferred to American, but Tea Rose and White Rose are found quite suitable as fuels; the ordinary Russian

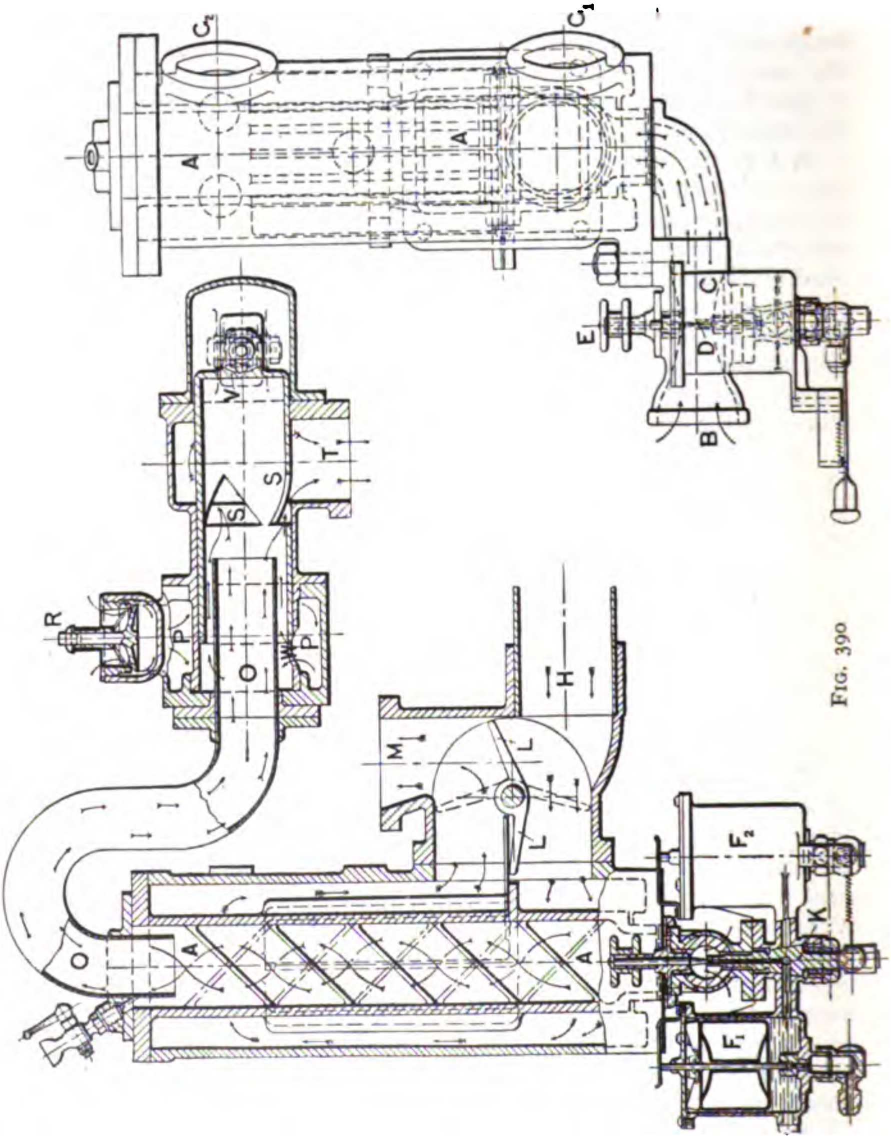


FIG. 390

brands used are Rocklight, Gladiator, Homelight, Phœbus, and Silver Spray. The Scotch 'Broxburn Lighthouse Oil' of sp. gr. 0·81 and flash point about 150° F., distilling between 300° F. and 570° F., may also be used ; for this heavy oil it is necessary to specially adjust the float and spray nozzle ; the engine is started on petrol as usual, but for lamp starting it is necessary to very thoroughly heat the vaporising tube and even part of the inlet pipe to prevent condensation in the mixture before the engine has become thoroughly warm.

In fig. 463 of Chap. XI an illustration is given of a four-cylinder, 6 ins. × 8 ins. Thornycroft marine paraffin engine showing the vaporiser just described *in situ* ; it will be noted that it is placed vertically ; the float feed box, with drip trough below, and air inlet are also clearly visible.

Exhaust-heated Vapourisers : General.—Exhaust-heated vaporisers operating on the suction-feed principle quickly respond to a governor-controlled mixture supply ; a spray of water can also be easily injected into the mixture in proportion to the air admitted, thus enabling higher compressions to be used and preventing the heavy knocking which otherwise is frequently found to occur in heavy oil engines when running at full load ; the water used should be as pure as possible.

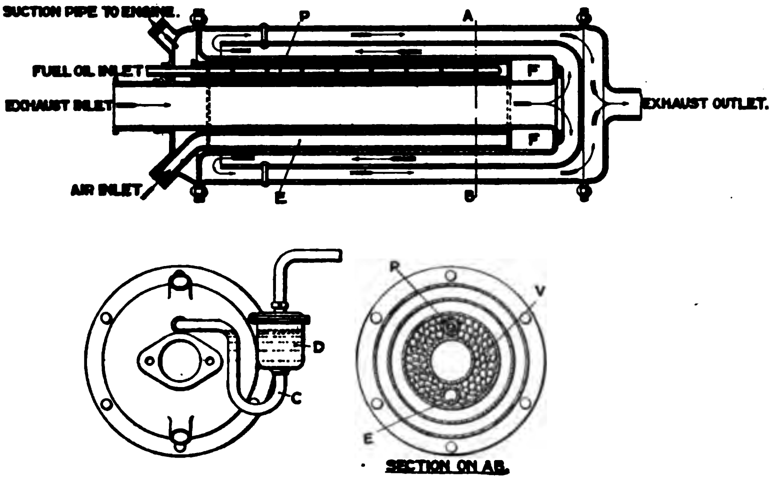


FIG. 391

This type of vaporiser is also at once applicable to the ordinary high-speed motor of automobile type as no alteration is necessary in the construction of the engine ; one vaporiser suffices for any number of cylinders of four-cycle type, and starting is conveniently effected, where permissible, by running the engine for a few minutes on petrol until the vaporiser and other parts are thoroughly warmed up ; the change from petrol to the heavier fuel can be instantly made by a two-way cock.

One of the most recent examples of this type is that of Mr. G. Constantinescu known as the 'G. C. Vaporiser,' diagrammatic views of which are given in fig. 391.

The device comprises a combined vaporising chamber and silencer, the exhaust gases from the engine being caused to circulate completely round the cylindrical annular vaporiser as indicated; the vaporising chamber V is loosely filled with blocks of rough cast iron or other suitable heat-retaining material, and the fuel oil is introduced into the perforated pipe P by way of the U-tube C, the quantity automatically varying with the engine suction by aid of the constant level float-feed device D; it will be noted that there is no spraying of the fuel oil charge, which merely drips through the perforations in the pipe P and wets the surface of the heated cast-iron blocks, thus becoming at once vaporised. Air is induced as indicated in the upper sectional view, and becoming heated in its passage through the pipe E enters the space F F, and passing back through the perforated diaphragm takes up the vapour rising from the cast-iron blocks, the resulting rich mixture then issuing by the suction branch shown. Between this suction branch and the engine are introduced a hand-controlled throttle valve and 'extra air' inlet device, either automatic or hand-controlled, of any ordinary simple design; the mixture is thus reduced in richness, and cooled, before entering the cylinder.

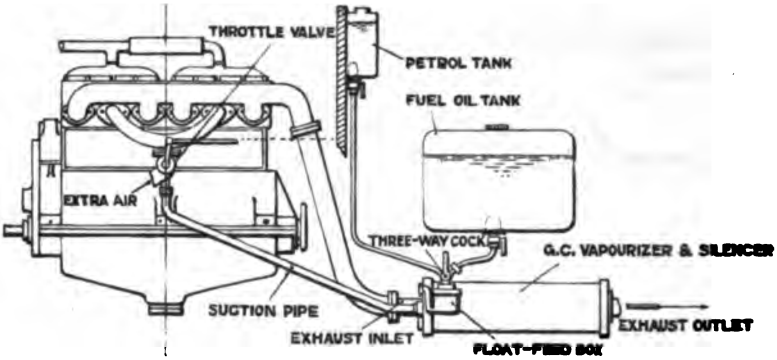


FIG. 392

To allow sufficient time for the complete vaporisation of the charge of fuel oil the dimensions of the vaporiser are so taken that the velocity of the rich mixture through it is only about one foot per second; the temperature of the vaporiser is usually from 550° F. to 750° F.; by the subsequent dilution with extra air the temperature of the mixture entering the engine is reduced to about 125° F. in ordinary cases. The general arrangement as applied to an automobile vehicle is shown in fig. 392; starting may be effected by blow-lamp heating the vaporiser silencer, but the more usual method is to fit an auxiliary petrol tank and start and run on this fuel until everything is well warmed up, when the heavy oil fuel can be turned on by the three-way cock shown; it is desirable to lag the suction pipe connecting the vaporiser with the engine.

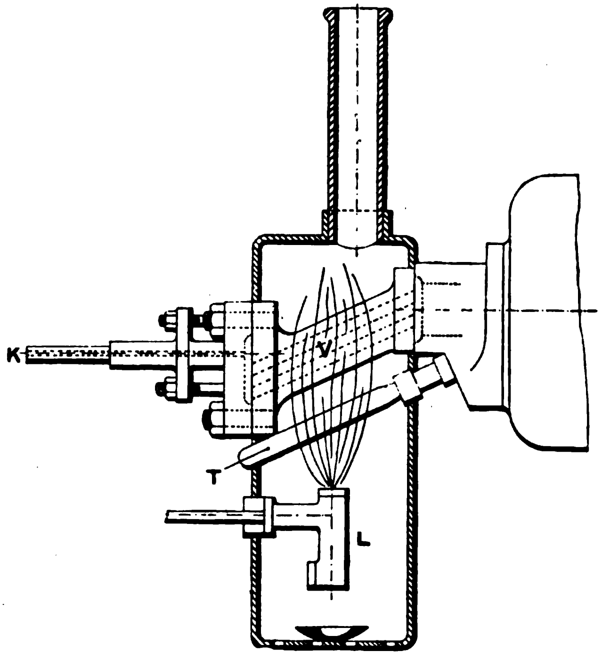


FIG. 393

The device has now (1912) been in use for a few months in certain commercial automobile vehicles and has produced a favourable impression; engines are found to run clean, and although not quite so controllable when using heavy oil as when using petrol are yet sufficiently so for the ordinary requirements of motor lurry service. As fuel, inferior burning oils and intermediate oils, costing about fourpence per gallon in quantity, are used; the consumption is, by bulk, said to be about the same as with petrol.

Type a_2.—Engines having lamp-heated vaporisers more nearly resemble in appearance the ordinary gas engine, and governing is very usually on the 'hit-or-miss' system; the vaporiser is generally a small retort-like chamber, and hot-tube ignition is frequently used, the same blow-lamp serving to heat both vaporiser and tube.

In the more recent Gardner designs the vaporiser is in one with the cylinder casting and consists, in fact, of the space below and around the inlet valve; ignition is here by low-tension magneto. The Gardner engine is fully illustrated and described in the succeeding Chap. XI. These later designs are thus included in 'Type b_2.'

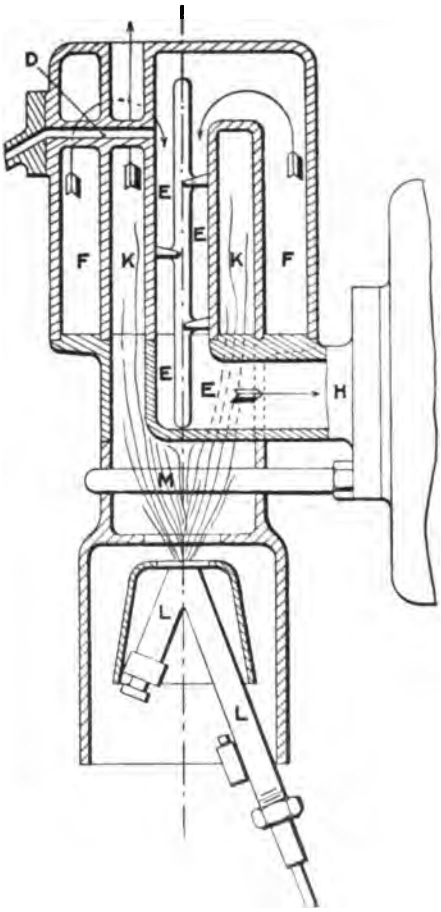


FIG. 394

Howard Vaporiser.—The 'Howard' lamp-heated vaporiser is diagrammatically illustrated in fig. 393; K is the kerosene supply pipe and V the vaporiser heated by the blow-lamp L, which serves also to keep the ignition tube T red-hot; the whole apparatus is small and compact, and is bolted to the end of the combustion chamber.

The Smith-Dudbridge Lamp-heated Vaporiser with its oil measuring cup device is illustrated in figs. 394 and 395; a fuel pump delivers oil through the nozzle A (fig. 395) and fills the cup B, the capacity of which is adjusted by the 'weir' control cock C C; P is an overflow pipe by which any excess of oil is returned to the reservoir. The measuring cup B is emptied at each induction stroke of the engine, the oil being sucked along the duct D into the lamp-heated vaporiser E E E E (fig. 394), where it is mixed with hot air from the outer jacket F F, and passes on to the cylinder by way of the branch H. K K is the lamp chimney surrounding the vaporising chamber E, and itself surrounded by the hot-air jacket F. The blow-lamp L L serves to heat both the vaporiser and the ignition tube M.

Crossley Bros. Oil Engines.—Messrs. Crossley Bros. build oil engines with (*a*) lamp ignition, (*b*) lampless ignition, and (*c*) a special design, also lampless, for crude and residual oils. In the lamp type engines the arrangement adopted is indicated in sectional elevation and plan in fig. 396; the blow-lamp A serves to heat both the ignition tube B and vaporiser C. During the suction stroke of the piston a small quantity of air is drawn into the vaporiser, becoming heated by passing around the lamp chimney jacket D D; simultaneously the charge of oil is sucked into the vaporiser from the syphon oil measuring cup E, of which a section in more detail is given in fig. 397; the rich mixture thus obtained is gasified in C, and enters the cylinder by way of the vapour valve F, where it mingles with fresh air admitted through the main inlet valve H. K is the exhaust valve. During compression the valve F is closed and the fresh charge packs into the firing tube B, which is so adjusted as to cause explosion to occur at the right instant.

The lamp-heated kerosene engines are recommended for adoption in cases where long runs at light loads occur. Referring to fig. 397, which shows the Crossley oil-measuring device, A is the oil delivery pipe from the oil pump by which the V-shaped 'syphon' cup B B is filled, any excess overflowing and returning by way of the pipe C to the oil reservoir. D is a set screw by which the capacity of the cup is regulated. E is the branch communicating with the vaporiser, through which the oil passes from B on the suction stroke of the piston, through a small non-return valve F carried in the screwed plug G. The Crossley lampless engines are of type b_1 with hot bulb and automatic tube ignition, and are illustrated and referred to later in this chapter.

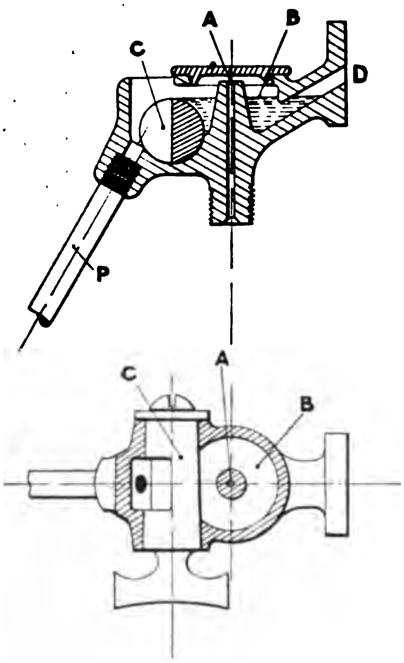


Fig. 395

Type a_3: Moorwood-Bennett Kerosene Carburettor.—Of the few exhaust-heated kerosene carburettors, the Moorwood-Bennett may be referred to as a typical example; the device resembles the early 'bubbling' petrol carburettors with the addition of an exhaust-heated jacket. It is illustrated in fig. 398; air enters through the sleeve controlled inlet A A, and passing down the central tube, bubbles up through the heated kerosene contained in a second concentric tube B B, which is surrounded by a third tube C C, through which the exhaust is passed. The rich mixture of vapour and air passes up through the ring of holes near the top of the second tube into the chamber D, and thence to the engine by the branch E; additional air is admitted between E and the cylinder in order to sufficiently reduce the richness of the mixture. The exhaust enters and leaves at F F; L is a drain cock and R the mixture regulating valve; K is the kerosene supply branch and H a small pipe in communication with the air.

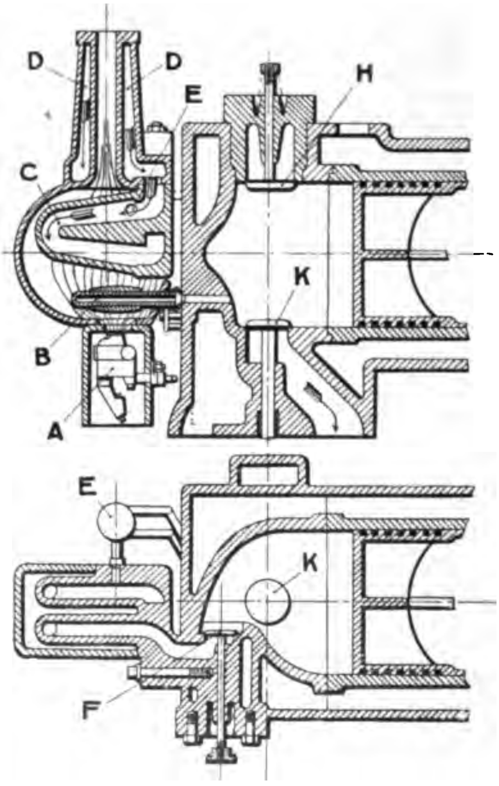


FIG. 396

Carburettors of this type, though often tried, have not hitherto met with much success in practice.

Type b_1: Hornsby-Akroyd Heavy Oil Engine.—Of oil engines having a combined vaporiser and explosion chamber forming part of the cylinder head, the Hornsby-Akroyd is one of the earliest and most widely known.

Invented by Mr. Akroyd Stuart (1886–1890), it was the first to successfully utilise the heated walls of a special portion of the combustion chamber in order to vaporise the oil fuel and also ignite the resulting working charge.

In fig. 399 a sectional view is given showing the cylinder and 'hot-bulb' vaporiser of the Hornsby-Akroyd oil engine; the vaporiser is so arranged that the heat of the successive explosions maintains it at a temperature sufficiently high to vaporise the oil by mere injection upon its hot surface, the heat also sufficing to produce ignition of the mixture at the completion of compression. The vaporiser is heated for 5 to 10 minutes by a blow-lamp in order to start the engine; the flywheel being turned, the piston on its suction stroke induces a charge of air into the cylinder, this air entering the cylinder direct, without passing through the vaporiser; at the same time the oil that has been injected into the hot bulb is vaporising, and diffusing through the chamber, being mixed, however, only with the hot exhaust products remaining from the preceding cycle. On the compression stroke of the piston the air passes through the narrow neck into the vaporiser, and there mixes with the oil vapour. The mixture is at first too rich to ignite, but the engine is so adjusted that just as compression is completed proper explosive proportions are attained by the vaporiser contents; the heat of the walls then causes ignition, and the piston moves outward and performs its working stroke under the pressure resulting from the explosion.

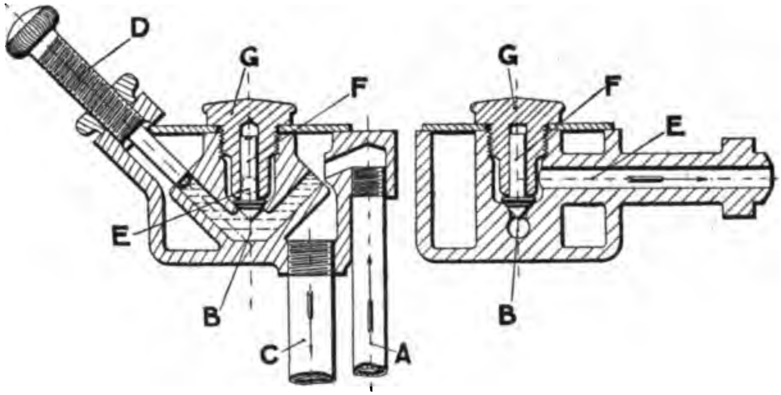


FIG. 397

The engine thus operates on the four-cycle, single-acting principle.

It is important that the vaporiser be not allowed to become too hot, as in this case pre-ignition occurs, and also the oil may become 'cracked,' with resulting deposition of carbon, which in time chokes up the vaporiser. In small engines the vaporiser and neck are not water cooled, but in the larger sizes it is found necessary to fit regulated water cooling to the neck and even part of the vaporiser itself, as indicated in the sectional view, fig. 399.

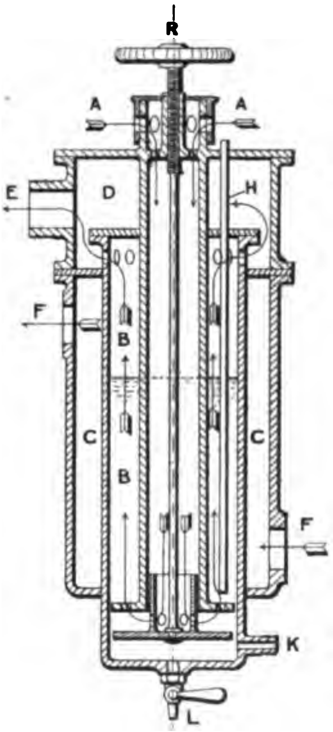


FIG. 398

The necessity of thus jacketing the neck, &c., was realised by Mr. Akroyd Stuart as the result of early experiences, and the regulation of the vaporiser temperature by means of jacketing formed the subject of one of his patents.

Some remarks upon the behaviour of oil vapour mixed with air and in contact with hot metal surfaces, together with a reference to the Clerk bolt igniter, will be found in Chap. III of this volume, dealing with methods of ignition (pp. 278 *et seq.*).

Several tests of the Hornsby-Akroyd engines were made between 1891 and 1898 by Professor Robinson with various fuels, including an intermediate oil of 0·853 sp. gr. with the high flash point of 225° F. The oil usually employed was, however, H.V.O. (heavy vaporising oil), or 'Russolene' having a sp. gr. of about 0·824 and flashing at 88° F.; this oil consists of 85·94 per cent. carbon and 14·06 per cent. hydrogen, and has a (higher) calorific value of 19,900 B.Th.U. per lb. One trial by Professor Robinson of a 5 BHP engine of 8·02 ins. bore × 14 ins. stroke, running at 214 revolutions per minute, using Russolene as fuel, furnished the following results at full load:

Revolutions per minute	214·3
Explosions per minute	91·4
Indicated horse-power	6·07
Brake horse-power	4·95
Mechanical efficiency, per cent.	81·5
Oil per BHP hour, lbs.	1·04

Temp. of jacket water leaving cylinder	126° F.
Rise of temperature of jacket water	63·6° F.
Lbs. of jacket water per minute	7·33

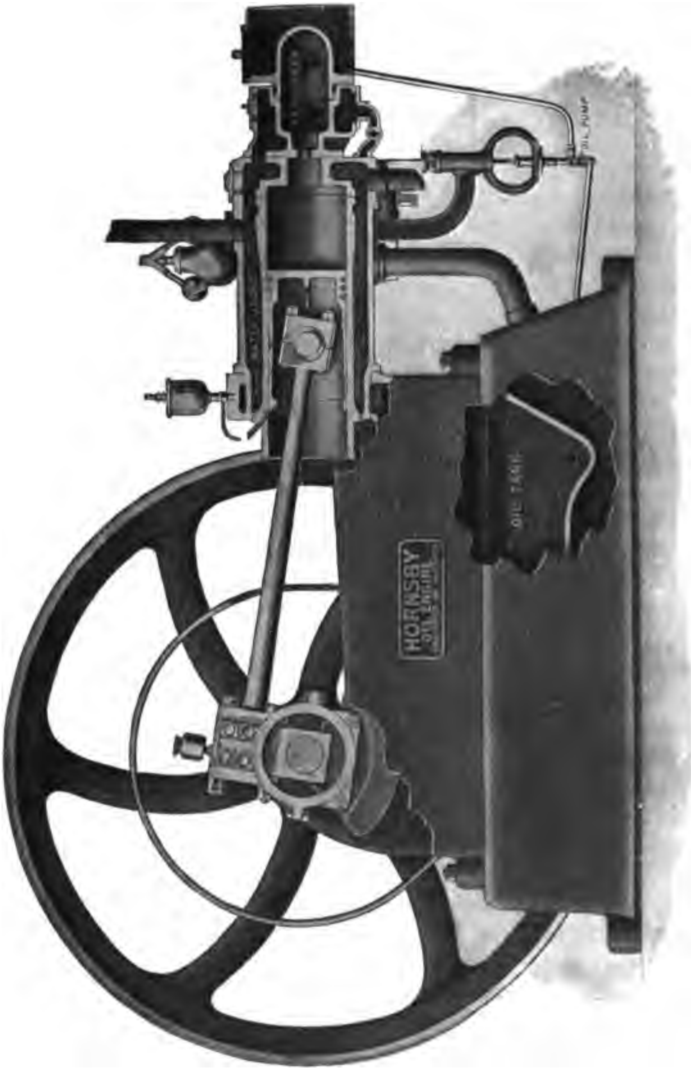


Fig. 399

Actual compression pressure, lbs. per sq. in. above atmosphere	43·0
Actual explosion pressure, lbs. per sq. in. above atmosphere	120·0
Mean effective pressure in lbs. per sq. in.	37·3
Percentage of heat of fuel appearing as IHP	16·1
Percentage of heat in jacket water	29·4
Percentage of heat in exhaust gases, radiation, &c.	54·5

The brake thermal efficiency was 13·1 per cent. In the estimation of these heat proportions Professor Robinson uses the lower heat value of the fuel, viz. 18,600 B.Th.U. per lb.

The temperature of the exhaust gas exceeded 600° F.; the exhaust consisted of 8·7 per cent. CO_2, 9·1 per cent. free oxygen, and 82·2 per cent. of nitrogen, by volume, in addition to condensed water; the proportion of air supplied to air burned was therefore 1·7 (*v.* Eq. (2), p. 726).

The piston speed of this small engine was 500 ft. per minute; Professor Robinson remarks that the range was from about 450 ft. per minute in the 1½ HP engine to 700 ft. per minute in the 50 HP size. He states further that too low a piston speed may give pre-ignition effects, and that the power and efficiency in these oil engines appear to increase together up to a piston speed of between 600 and 700 ft. per minute. In 1898 Professor Robinson tested a 25 HP engine of 14·5 ins. bore and 17 ins. stroke, again with Russolene as fuel; at full load the engine made 202·6 r.p.m. with 101·3 explosions, the BHP being 26·74 and mechanical efficiency about 84½ per cent. The oil used per BHP hour in this larger engine was only 0·74 lb. The compression pressure was 60 lbs. per sq. in. above atmosphere; explosion pressure 168 lbs. per sq. in.; and MEP about 44 lbs. per sq. in. The brake thermal efficiency was 18·5 per cent., and the indicated thermal efficiency 21·9 per cent., reckoned, as before, upon a (lower) heat value of 18,600 B.Th.U. per lb. of fuel. Russolene was found to permit a higher compression, and to give fully 15 to 20 per cent. more power, with better running, than Royal Daylight—with which pre-ignition occurred unless the compression pressure was reduced. The best compression pressure for use with any given oil is obtained by experience in actual practice. For their 2 to 4 HP engines Messrs. Hornsby use different vaporisers according to the oil employed; from 5 HP to 45 HP inclusive, in addition to using different vaporiser cap ends, a cover of conical form, containing a solid block, is bolted to the side of the vaporiser. The device is illustrated in fig. 400; No. 3 shows the normal mode of fitting; for oil requiring a higher compression No. 4 is the arrangement adopted; while for low compressions Nos. 5 and 6 are used.

For engines of and exceeding 55 BHP, in addition to using different vaporiser cap ends, Messrs. Hornsby also vary the compression ratio by fitting distance pieces to the big end of the connecting-rod. When using Russian oil these distance pieces are inserted between the rod end and the brass nearest the cylinder, thus increasing the compression.

In addition to kerosenes, these engines run satisfactorily on crude, gas, and residual oils; for the English market they are built for stock at about 86° F. by Abel close test.

The Royal Agricultural Society of England carried out a series of trials of oil engines at Cambridge in June 1894, the judges being Professors Ewing and Capper and Mr. J. B. Denison ; the engines were run at full load with Russolene for three days, and a special trial was then made at full power ; the Hornsby engine was awarded the first prize for simplicity, neatness of design, and regularity and steadiness in running.

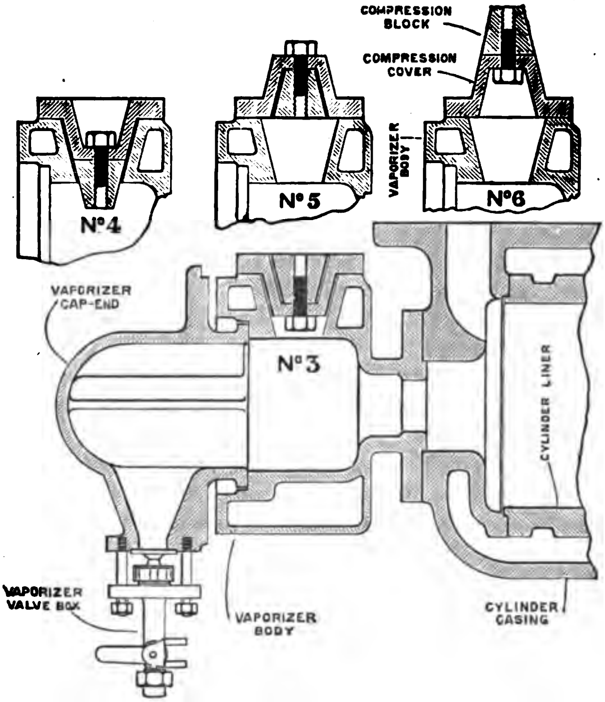


FIG. 400

The engine tested was given as of 8 BHP, and its dimensions were : diameter of cylinder 10 ins., stroke 15 ins., weight of engine 40 cwt. During the trials, according to Professor Capper's report, the engine ran without hitch of any kind from start to finish. Its action was faultless. One attendant only was employed all through the trials, and started the engine easily and with certainty after working the hand blast to the lamp for 8 minutes. During three days' run the longest time taken to start was 9 minutes, and the shortest 7 minutes. When the engine stopped each day the bearings were cool and the piston was moist and well lubricated; the revolutions were very constant, and the power developed did not vary one quarter of a BHP from day to day. The oil consumed, reckoned on the average of the three days' run, was 0·919 lb. per BHP per hour. The oil used was Russolene, sold in Cambridge at that time at the price of 3¼*d*. per gallon. At this rate the cost for oil per BHP hour was ½*d*., and this included also the oil used for the starting lamp.

Mr. C. F. Wilson, F.C.S., made an analysis of the Russolene oil used for the purpose of testing the oil engines exhibited at Cambridge, and found the specific gravity at 60° F. 0·824, the flashing point (Abel test) 88° F., and the total heat of combustion 11·055[1] calories, but after deducting for the heat due to the condensation of water vapour this reduced to 10·313[2] calories. The oil contained 14·05 per cent. hydrogen. Mr. Wilson makes the observation that this oil appears to be very constant in composition, as a similar oil examined by him a year before gave 14·07 per cent. hydrogen, and a corrected calorific value of 10·3 calories, so that the two samples supplied at an interval of a year were practically identical.

The mean power exerted during the three days' trials was 8·35 brake horse. At a subsequent full-power trial of the same engine at the show, a BHP of 8·57 was obtained, the engine running at a mean speed of 239·66 revolutions per minute and the test lasting for two hours; the indicated power was 10·3 horse, the explosions per minute 119·83, the mean effective pressure 28·9 pounds per sq. in.; the oil used per IHP per hour was 0·81, and per BHP per hour 0·977 lbs. According to Professor Capper the heat account of the engine was:

	Per cent.
Heat shown on indicator diagram per IHP	16·9
Heat rejected in jackets	29·5
Heat rejected in exhaust and other losses	53·6
	100·0

In these tests Professor Capper takes the corrected heat value of the oil instead of the total heat value. In determining the absolute efficiency of any engine, it is necessary to take as a basis the total amount of heat evolved by the combustion *from* the atmospheric temperature *to* the atmospheric temperature again. The author has accordingly recalculated these quantities, and finds the correct heat account below:

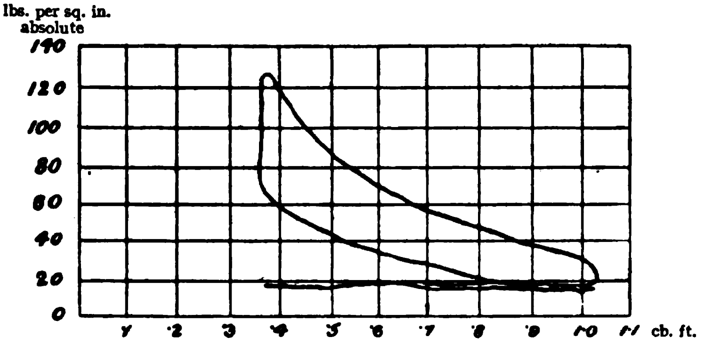
	Per cent.
Heat shown on indicator diagram per IHP	15·3
Heat rejected in jackets	26·8
Heat rejected in exhaust and other losses	57·9
	100·0

---

[1] 19,870 B.Th.U. per lb.

[2] 18,550 B.Th.U. per lb.

In a test of this engine made at the same time, but at half-power, the BHP developed was 4·57 at 235·9 revolutions per minute, and the oil used per BHP per hour was 1·49 lbs. On a four hours' test with this engine running entirely without load at 240 revolutions per minute it was found that it consumed 4·23 lbs. of oil per hour. Fig. 401 is a card from the Hornsby engine, being an average card of the two hours' full-power trial. The cylinder volume is given in cub. ft. and the compression space is also given. From this diagram it will be observed that the average pressure, the maximum pressure, and the pressure of compression are very low, and that consequently a large cylinder is required to develop a given power, while it is worth observing how beautifully regular is the ignition obtained by the simple device of firing from the surface of the hot combustion chamber.



Brake HP, 8·57; indicated HP, 10·3; diam. of cylinder, 10″; stroke, 15″; revs. per min., 239·66; explosions per min., 119·83; mean pressure, 28·9 lbs. per sq. in.; pressure of explosion, 112 lbs. per sq. in. above atmos.; pressure of compression, 50 lbs.; oil per IHP hour, 0·8 lb.; oil per BHP hour, 0·977 lb.

FIG. 401.—Hornsby-Akroyd Oil Engine (diagram). Average card, two hours' full-power trial. Russolene oil.

The table on the following page gives the general results of these trials.

A test in September 1908 made by Professor. W. Robinson, M.Inst.C.E., on a single-cylinder, horizontal, 32 BHP Hornsby engine furnished results as given below; the time occupied in starting from cold was ten minutes. The fuel used was Russolene, having a (lower) calorific value of 18,450 B.Th.U. per lb.

Duration of trial	1 hour
Compression pressure, lbs. per sq. in.	85
Explosion pressure, lbs. per sq. in.	260
Revolutions per minute, mean	230·2
Brake horse-power	32
Total oil burned, lbs.	19·6
Lbs. oil per BHP hour	0·613
Per cent. brake thermal efficiency	22·6

R.A.S.E. TRIALS OF OIL ENGINES. (*Cambridge.* 1894)

Name of engine	Hornsby Akroyd	Crossley	Premier	Trusty	Campbell	Britannia
Bore and stroke, ins. .	10 × 15	7 × 15	8¼ × 15	6¾ × 13	7½ × 12	7½ × 13
Clearance volume, cub. ins.	638	225·9	360·8	214·7	—	260
Weight of engine, lbs. .	4500	3640	4100	2900	3025	3700
Mean revs. per min. . .	240	201	160	260	208	240
Piston speed, ft. per min. .	600	500	400	565	416	520
Mean explosions per min. .	120	75·3	72·2	119·0	67·7	120·0
MEP in lbs. per sq. in. .	28·9	72·2	49·6	46·1	65·5	47·3
Indicated horse-power .	10·3	7·9	7·3	6·5	5·9	8·4
Brake horse-power . .	8·57	7·01	6·46	4·73	4·81	6·21
Mechanical efficiency, per cent.	83·3	88·8	88·6	73·0	81·6	74·0
Total oil, engine & lamp, lbs.	16·75	11·5	15·25	11·25	10·75	20·0
Lbs. of oil per IHP hour .	0·81	0·73	0·93	0·87	0·93	1·25
Lbs. of oil per BHP hour .	0·977	0·82	1·04	1·19	1·12	1·68
Duration of trial, mins. .	120	120	136	120	120	115
Lbs. of jacket water per min.	10·9	10·1	10·0	9·2	16·6	17.3
Rise of temp. of jacket water, ° F.	70	42	80	42	33	38
Final temp. of jacket water, ° F.	143	115	153	115	110	101
Brake thermal efficiency per cent.	13·1	15·6	12·3	10·75	11·4	7·6
FROM 3 DAYS' TRIAL :						
Brake horse-power .	8·35	6·28	5·96	4·63	4·75	6·15
Lbs. oil per BHP hour .	0·919	0·9	1·06	1·157	1·15	1·49

The brake thermal efficiency is estimated by aid of the lower calorific value of the fuel in this case.

The Hornsby oil engines are governed by a loaded 'Porter' governor connected with the fuel oil delivery to the cylinder in such manner that the amount of oil injected into the hot bulb is varied with the speed of the engine, the excess oil pumped returning to the reservoir ; no explosions are thus missed as in the 'hit-or-miss' system of governing, and the engines run very steadily ; the governor valve is illustrated in Chap. IV, fig. 251. An illustration is given in fig. 402 of one of the portable Hornsby oil engine arrangements ; these are constructed in sizes from 2 BHP at 400 r.p.m. to 12 BHP at 265 r.p.m. Thermo-syphon cooling is employed, the water being contained in the cylindrical galvanised tank shown. These portable plants are very useful for threshing and agricultural work generally.

For the fixed oil engines with gravity circulation of the cooling water, not less than 62 gallons per BHP should be provided in this country, and the temperature on leaving the top of cylinder should be more than 100° F. and less than 140° F. ; rain water should be used

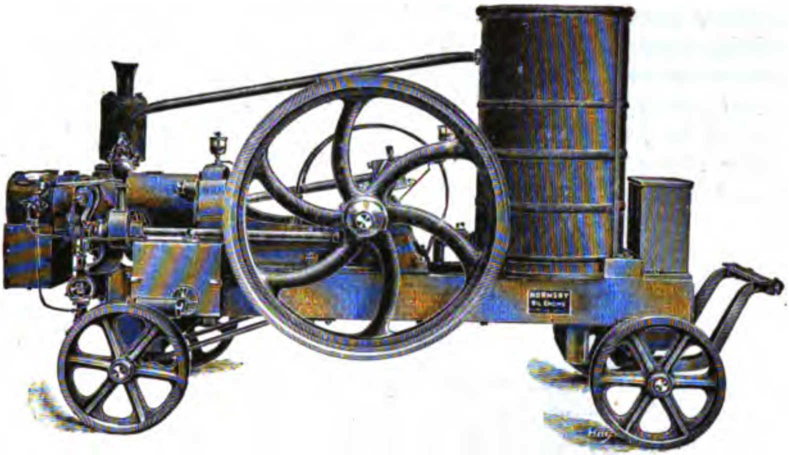


FIG. 402

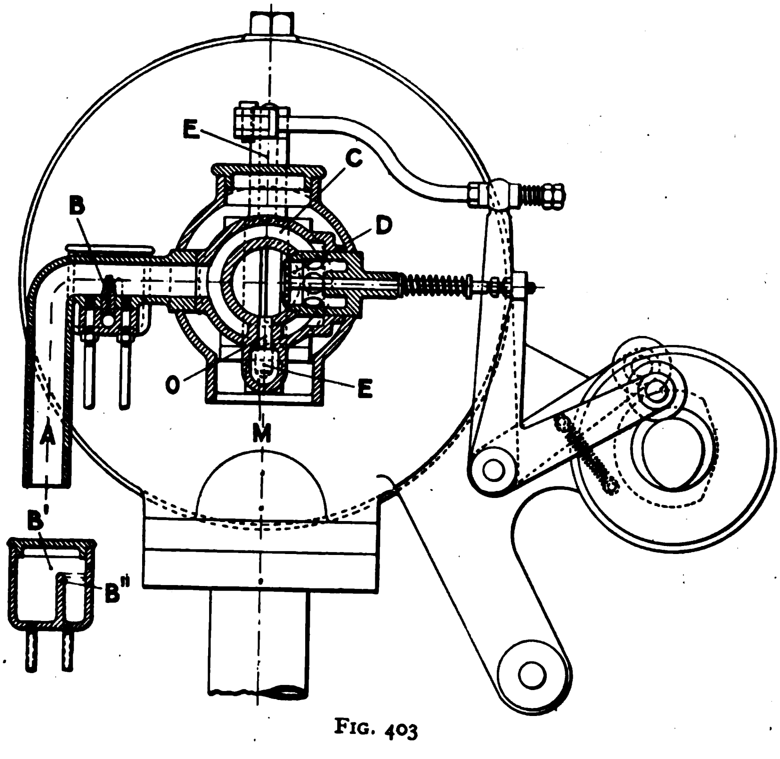


FIG. 403

wherever possible. The Hornsby oil engine is made in the single-cylinder type up to 185 BHP, and with two cylinders to 370 BHP. A test of a 100 BHP engine of 502½ consecutive hours' duration, using as fuel a heavy Texas oil of 0·933 sp. gr. and 240° F. flash (open test), resulted in a consumption per BHP hour of 0·578 pint only. The engine was rated as of 100 BHP with residual oil, or 110 BHP with refined oil (i.e. Russian kerosene)

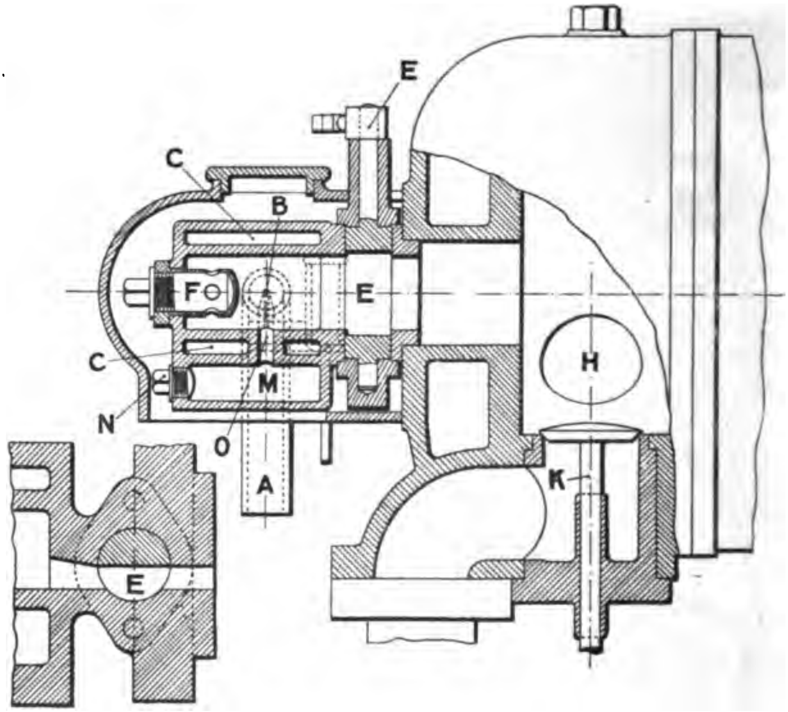


FIG. 404

The Blackstone Oil Engine.—Another well-known engine with a combined vaporiser and explosion chamber forming part of the cylinder head is the Blackstone kerosene engine, constructed in sizes from 2 to 70 horse-power.

The marine crude oil Blackstone differs from their kerosene type in many details; these crude oil engines are separately described in Chap. XI of this volume, on Marine Oil Engines. The arrangement adopted in the kerosene engines is shown in the sectional views, figs. 403 and 404. During the suction stroke of the piston air is drawn in through the 'inspirator' tube A, the current traversing the nozzle B and 'snifting' therefrom a charge of oil spray; the rich mixture thence passes into the vaporiser C, which is an annular chamber or jacket surrounding the heated explosion chamber, where the spray is gasified. From the vaporiser it passes by way of the mechanically operated vapour inlet valve D into the explosion chamber. E E is the ignition timing valve, which closes at the end of the suction stroke and reopens near the end of the compression stroke, thus admitting the compressed charge into the explosion chamber, where it is fired by an automatic ignitor F, consisting of a flat coil of wrought iron contained within a perforated casing. The ignitor projects inwards from the centre of the back end of the explosion chamber and is maintained at a red heat by the successive explosions. H is the main air inlet valve into the combustion chamber, and K is the exhaust valve.

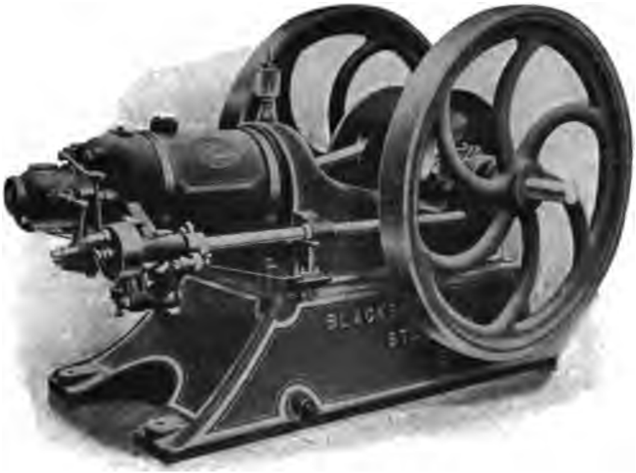


FIG. 405

The timing valve E E is a cylindrical plug valve half cut away as shown in the enlarged section fig. 404, so as to give a narrow rectangular passage of communication between the explosion and combustion chambers; the plug is made a quite easy fit in its casing and lasts well in service. The instant of opening is easily regulated, and the moment of ignition thus readily adjusted; Messrs. Blackstone state that they find water spray injection into the cylinder unnecessary with this arrangement.

The spray nozzle B is in communication with a small oil-cup B′, which is maintained full to overflowing by a little plunger pump driven from the engine; any excess oil pumped overflows the weir B″ and returns to the oil reservoir.

Running along the bottom of the explosion chamber and connected always with it through the passage O is a parallel cylindrical small chamber M, closed at the rear end by a screw plug N. To start the engine this chamber is blow-lamp heated for about five minutes, and the first few explosions are caused by contact of the compressed charge with its heated surface; as soon as the temperature of the igniter F is sufficiently raised, the lamp may be withdrawn.

When the engine is required to run for prolonged periods at small loads Messrs. Blackstone fit an additional auxiliary igniter in the chamber M, attached to the end-plug N. For occasional light load periods the lamp is sometimes desirable in order to keep the vaporiser sufficiently hot and prevent a smoky exhaust.

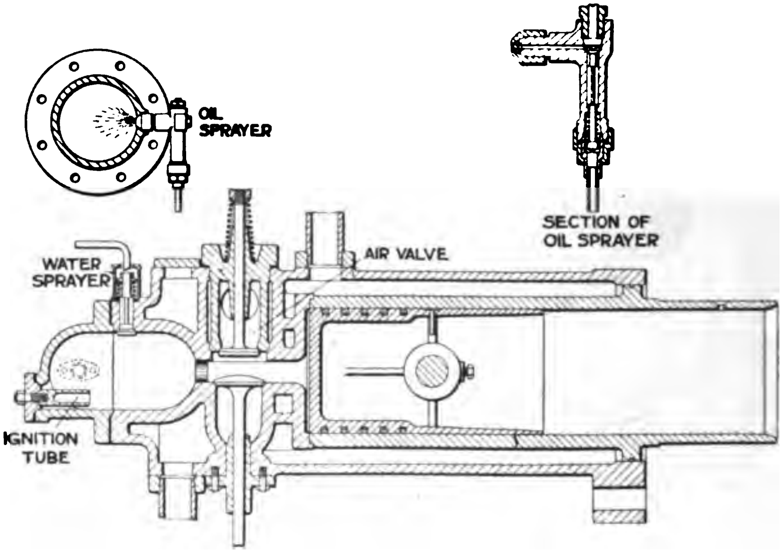


Fig. 406

The engine is simple and neat in external appearance; a general view of the type with oil tank foundation is given in fig. 405. Any ordinary refined oil, as Rocklight, Russolene, Royal Daylight, &c., is used.

The Crossley Crude Oil Engine.—In the Crossley lampless oil engines, suitable for use with crude and residual oils, vaporisation is also effected by injection of the oil direct into a hot bulb forming a prolongation of the combustion chamber; ignition is automatically effected by a hot tube projecting into the explosion chamber and maintained at the necessary high temperature by heat from the successive explosions. A sectional view is shown in fig. 406.

This requires no detailed explanation; it will be observed that the very usual water spray injection is employed.

The Robey Oil Engine.—In the early Norris vaporiser the hot bulb projected inwards into the combustion chamber, as illustrated diagrammatically in fig. 407. It will be noted that the combustion chamber is here well water-jacketed. The fuel oil is injected by a small force-feed pump through the branch A into the internal hot bulb B contained within the combustion chamber CC. Fresh air enters at E during the suction stroke, and passing the main air valve F is delivered directly into the cylinder D D as indicated by the arrow; on the return stroke the air is compressed into the combustion chamber and hot bulb B containing the vaporised oil; the proportions of air and oil vapour are so adjusted that the mixture is automatically ignited by the bulb at the proper moment. The burnt gases all pass through the bulb, the exhaust valve H being placed at the extreme rear of the engine; the heat of the bulb is thus maintained and its surfaces said to be kept clean and free from tarry deposit. The bulb is heated at starting by a special blow-lamp; in the earliest types used by Messrs. Robey the lamp was supplied with an air blast produced by a hand-driven fan, but in later practice an automatic blow-lamp was employed.

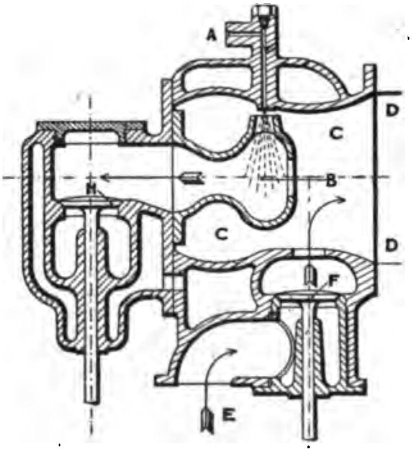


FIG. 407

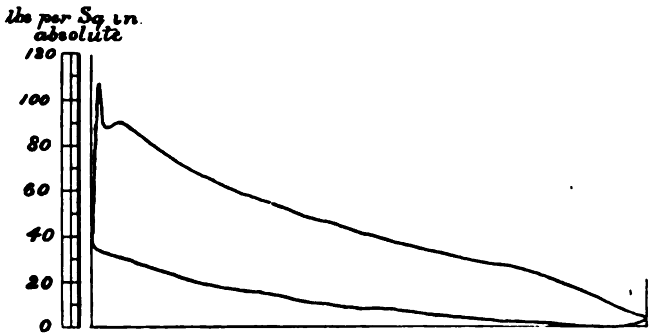


FIG. 408.—Robey Oil Engine (diagram, Clerk)

Fig. 408 shows a diagram taken by the author from a Robey oil engine having a vaporiser of this type; the bore was 6 ins. and stroke 9 ins.; the engine ran at 260 r.p.m. As fuel an American oil of sp. gr. 0·857 at 50° F. was used. The vaporising and ignition of this engine appeared to be effective and regular.

In the early Capitaine oil engines an arrangement substantially similar to that just described was used. The combustion chamber was water-jacketed and the oil was injected centrally into an internally projecting heated vaporiser and igniter combined. In later designs of the Capitaine engine an external lamp-heated vaporiser of type b_2 is used; this is referred to later.

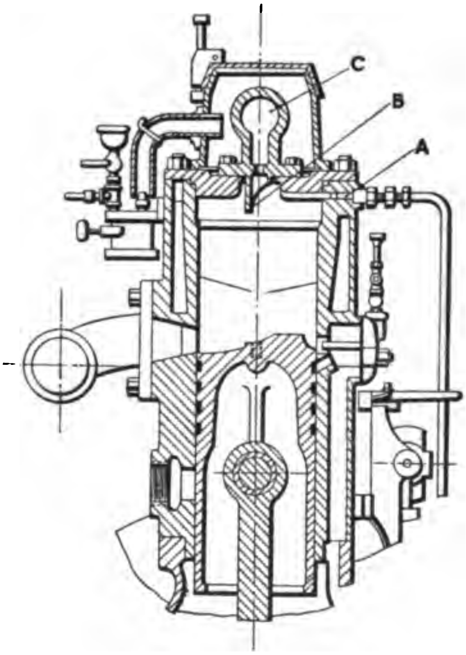


Fig. 409

The Mietz & Weiss Oil Engine.—In the well-known American 'Mietz & Weiss' oil engine, which operates on the usual two-stroke cycle with crank-chamber air compression, external automatic hot-bulb ignition is employed; the general arrangement is indicated in fig. 409. The oil is sprayed from the inlet A upon the lip B, extending into the combustion chamber from the passage communicating with the hot bulb C. Provision is made for the introduction of a small quantity of steam—obtained from the jacket water—into the working charge, which produces the usual 'softening' effect at full load running.

The Petter Oil Engine.—In the four-cycle Petter oil engine, introduced in 1896, ignition is by internal hot tube and is automatic, a blow-lamp being necessary only to heat the vaporiser at starting. By the governor the supply of oil and air to the vaporiser is simultaneously controlled so as to preserve an approximately constant mixture richness of diminishing *quantity* at light loads. This involves lessened compression and less economical consumption at such times. Water injection is employed to soften the running at full load.

The revolution speed of the Petter engines is comparatively high, though on account of the medium length of stroke the piston speed is about normal; short strokes and quick revolution speeds have characterised the Capitaine oil engines from the first; these were among the earliest to use ordinary kerosene with automatic ignition.

The Petter four-cycle engines, in sizes over 3½ BHP may be run on crude and residual oils, in addition to kerosene, with a small adjustment of the oil injection valve and air supply. The following table illustrates the Petter Co.'s practice with standard horizontal engines:

Horizontal Petter Oil Engines

B.H.P.		Bore in inches	Stroke in inches	Revs. per minute	Piston speed, ft. per min.	Weight complete in lbs.	Weight per normal BHP in lbs.
Normal	Maximum						
1½	2¼	—	—	450	—	670	445
2¼	3	4½	6½	450	490	780	346
3½	5	5¼	9¾	350	570	1680	480
6	7½	6½	9¾	330	540	2000	334
8	10	7¼	11½	270	520	3150	394
10	12½	8¼	11½	250	480	3600	360
15	18	9¾	15	230	575	5150	344
20	24	10¾	15	230	575	6000	300
26	32	12	17	230	650	9400	360
38	45	14	17	225	640	12,700	335
50	60	15	20	210	700	20,000	400

'Weight complete' includes the engine and all accessories, excepting only the water cooling tank and pipe connections.

Tests of an 8 BHP and of a 15 BHP Petter engine, each of one hour duration, and using Rocklight oil of sp. gr. 0·815, and flash point 78° F., made by the Petter Co., furnished results as follows:

Item	8 BHP engine				15 BHP engine			
	Maxm. load	Normal	Half	Light	Maxm. load	Normal	Half	Light
BHP .	10·0	8·0	4·5	0·0	18·0	15·0	8·52	0·0
Revs. per min.	261	262	264	266	218	218	222	224
Oil per hour, pints .	6·79	5·88	4·1	2·75	12·2	10·2	6·2	4·0
Oil per BHP . hour, pints .	0·679	0·735	0·913	—	0·68	0·68	0·73	—

An external view of the 38 BHP Petter oil engine is given in fig. 410. The Petter Co. apply their oil engines also to portable and traction designs with considerable success.

Type b_2: The Ruston Oil Engine.—In the Ruston oil engine the vaporiser is an unjacketed extension from the cylinder head, its temperature being maintained by the successive explosions after a preliminary heating by blow-lamp. The general arrangement is illustrated in fig. 411; the oil fuel is delivered by a pump from the engine-bed reservoir to a small cistern (see fig. 412), whence a minutely regulated small quantity passes at each suction stroke of the piston, by way of the needle valve K, into the vaporiser V, which is a taper plug having a spiral groove, and seated in a casing surrounded by the belt B in communication with the combustion chamber. The charge of oil is vaporised in the spiral groove, and the rich mixture of vapour and air from V passes through the vapour valve O into the space B, and thence into the combustion chamber C, where it is mixed with fresh air entering through the main air valve. On the compression stroke the now inflammable mixture is forced back into the belt B, and is automatically ignited at the proper moment by the hot tube T.

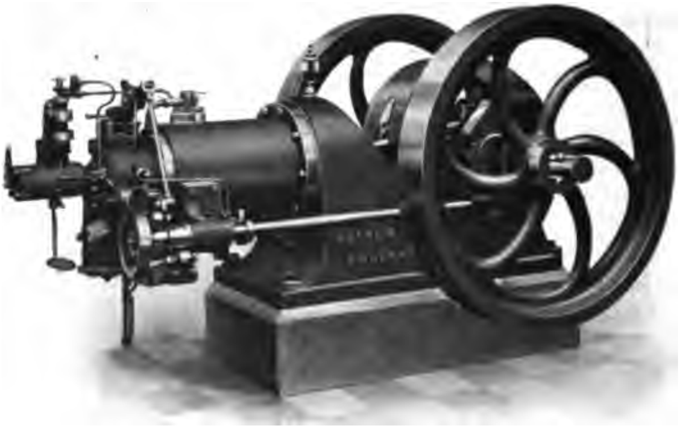


FIG. 410

The ingenious Ruston oil measuring cup is illustrated in fig. 412 in enlarged section. A is the kerosene supply from the pump; B is the oil cistern with weir C; D is the overflow return to reservoir; E is an annular slider, or bucket, operated by the cam shaft through gear F F; H is the supply pipe to the vaporiser.

The weir C fixes the level of the oil in the cistern, while the volume of the cup-shaped depression in the top of the annular slider determines the exact quantity of oil admitted at each suction stroke to the upper open extremity of the vaporiser supply pipe H.

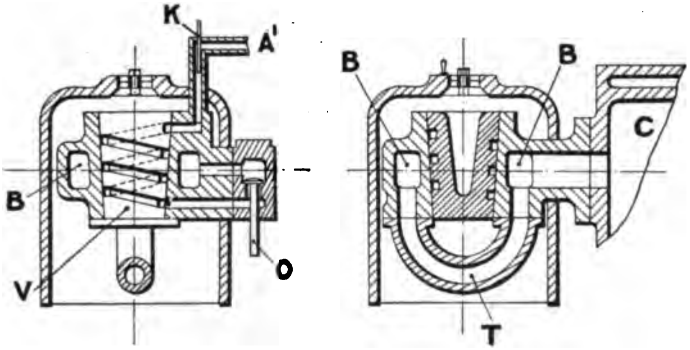


FIG. 411

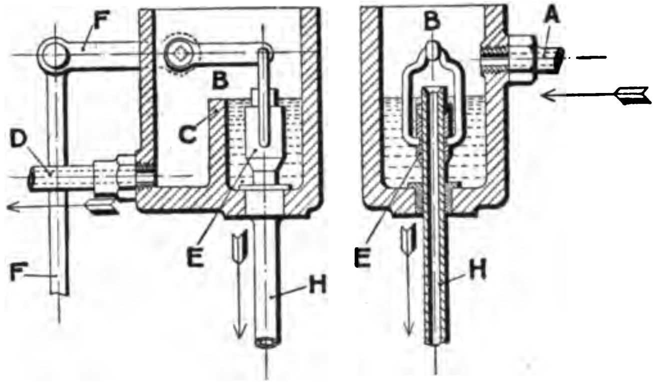


FIG. 412

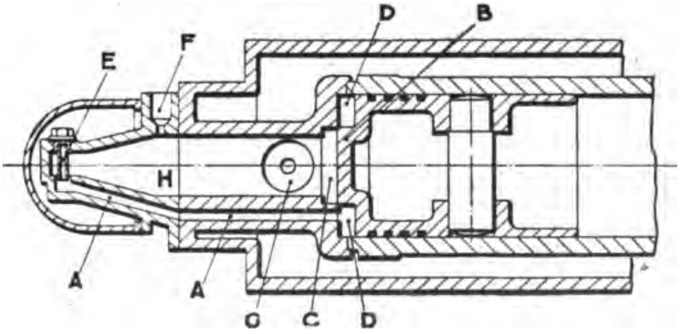


FIG. 413

These engines are built of the horizontal, single-cylinder type in sizes from 3 to 90 BHP inclusive; they may be run on refined oils (kerosenes), as Russolene, Rocklight, Homelight, Royal, Daylight, &c., and also on good crude oil, as Russian (sp. gr. about 0·89), and on inferior crude oils, as Borneo and Texas (sq. gr. about 0·925); the power developed in the three cases being about in the proportion 1, 0·8, 0·7 respectively. As usual, the Russian kerosenes are found to give somewhat better results than the American.

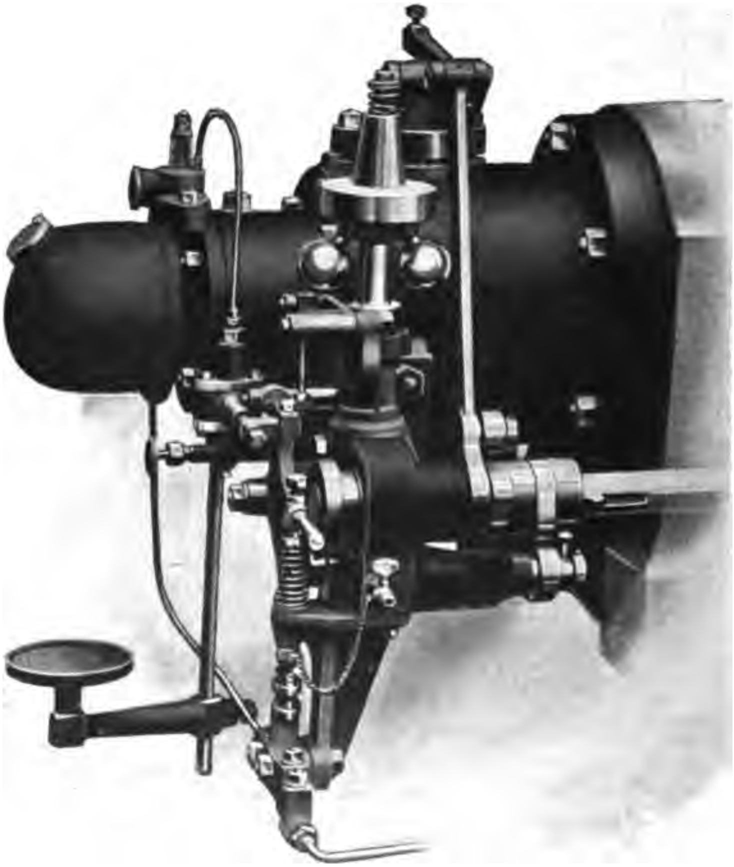


FIG. 414

It is recommended that for continuous working during periods exceeding about two hours the output should not exceed 90 per cent.

of the rated BHP. An account of the recent Ruston crude oil engine, together with some trial results, will be found near the end of this chapter.

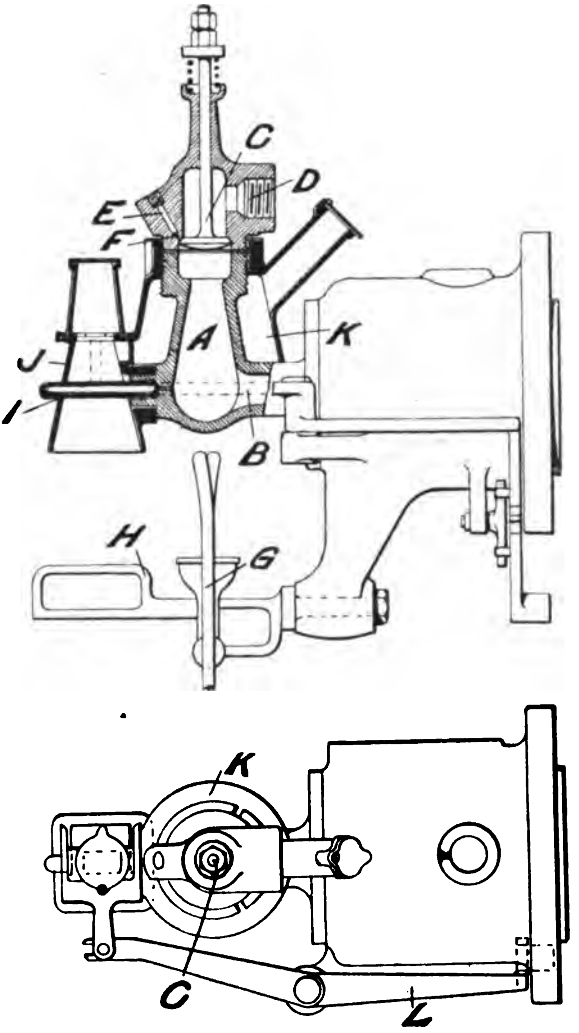


Fig. 415.—Tangye Oil Engine

The National Gas Engine Co.'s Oil Engines.—In the oil engines of the National Co. the vaporiser is an unjacketed extension of the combustion chamber as indicated in the accompanying fig. 413. A blow-lamp is needed only to heat the vaporiser at starting. The combustion chamber is water cooled, and of considerably smaller diameter than the cylinder, while the vaporiser is conical in form. A cylindrical projection, B, upon the crown of the piston enters the space C near the end of the stroke, with the result that the inflammable mixture entrapped in the annulus D D is forced to pass at considerable velocity along the small passage A A, at the end of which it is ignited by the incandescent tube E; the flash passes into the compressed mixture in the chamber H, and crisp and regular firing is thus ensured. The ignition tube is maintained at a sufficient temperature by the heat communicated by the successive explosions, and the firing is thus entirely automatic; further, the risk of pre-ignition is minimised by the use of the by-pass passage A A as described. The oil enters at F; G is the main air valve. Water injection is employed to secure quietness of running at three-quarter load and over; governing is on the hit-or-miss system. An external view of the cylinder end, showing the vaporiser, governor, oil-pump, lamp bracket, valve drive, &c., is given in fig. 414. The National Co.'s oil engines are built of the horizontal single-cylinder type in units of from 2 to 40 BHP, running at from 350 to 230 r.p.m.; kerosene, crude oils, and many 'residuals' may be used as fuel, the power obtained in the latter cases being only 85 to 90 per cent. of that when kerosene is employed.

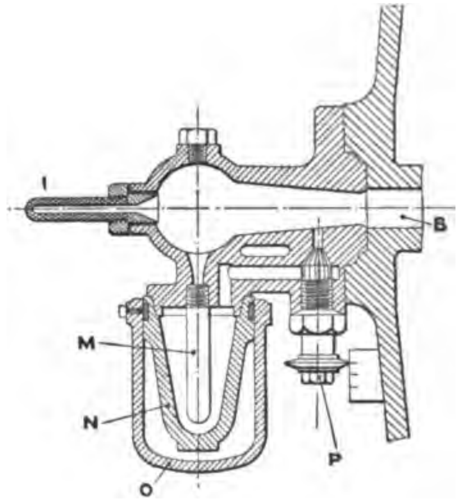


FIG. 416

The Tangye Oil Engines.—In the earlier Tangye kerosene engines lamp heating was employed to maintain the temperature of the vaporiser and ignition tube, the general arrangement being as indicated in fig. 415.

The bottle-shaped vaporiser A was in constant communication with the cylinder combustion chamber through the passage B. The kerosene supply branch E terminated in a fine duct F, opening in the seat of the automatic spring-controlled air inlet valve C. On the suction stroke of the piston, air, entering at D, passed the valve C, taking up a sprayed charge of oil from the orifice F in the usual manner of the 'mixing valve'; this mixture was gasified in the vaporiser, and at the end of the compression stroke fired by the ignition tube I. To start the engine the vaporiser was first heated by the blow-lamp G; the lamp was next moved to the bracket H, quickly raising the ignition tube I to incandescence; the engine was then started by turning the flywheel.

Governing was effected by holding up the exhaust valve, so that the exhaust gases were alternately drawn into and expelled from the cylinder; at such times, owing to the absence of any suction, the mixing valve C remained closed, so that neither oil nor air then entered the cylinder.

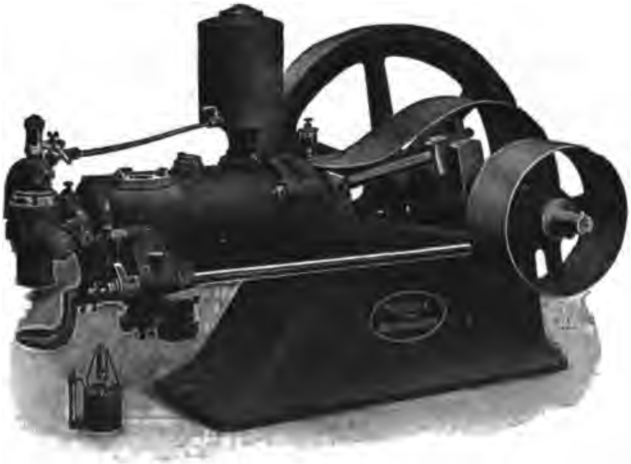


FIG. 417

In the present (1911) Tangye kerosene engines the blow-lamp is used only for starting purposes, the vaporiser and external ignition tube being thus heated; the engine runs on the lamp-heated external ignition tube until thoroughly warmed up, when the lamp is removed, ignition thereafter being caused automatically by a second ignition tube placed at the side of the vaporiser.

Fig. 416 illustrates the arrangement adopted. I is the external ignition tube as before, while M is a second tube whose temperature is maintained by being surrounded with hot gas contained in the cap N; the admission of this gas from the vaporiser is hand-regulated by the small screw valve P. The temperature of the cap N and its contents is preserved by an outer cover O.

An external view of the present Tangye kerosene engine is given in fig. 417, the automatic ignition tube cover projecting from the side of the vaporiser being clearly shown. A timing valve is now employed to determine the instant of firing.

Excepting only in the smallest sizes, water injection into the cylinder is employed in order to obtain economical and quiet running. These engines are built of the horizontal, single-cylinder type from 1½ to 45 BHP. Messrs. Tangye construct also a 'crude oil engine' with automatic hot-bulb ignition. This engine works on the ordinary four-stroke cycle, and is governed by varying the quantity of oil injected into the vaporiser by the feed pump. Water injection is employed to regulate the temperature of the vaporiser at all loads.

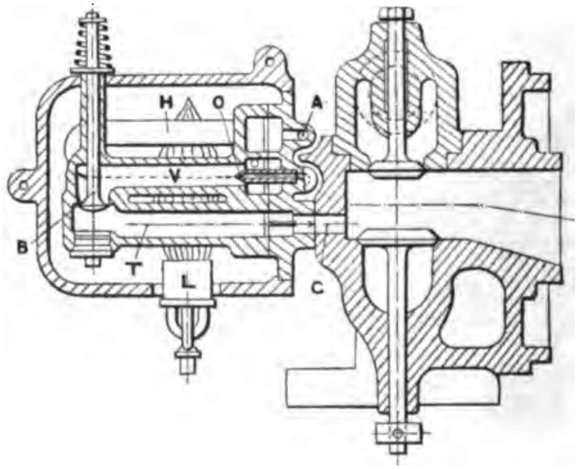


FIG. 418

The Fielding & Platt Oil Engines.—Messrs. Fielding & Platt also used in their first oil engines a lamp heated arrangement in which the vaporiser and ignition tube were combined. The device is shown diagrammatically in fig. 418. Air entered at A, became heated in passing through the U-tube H, and, mixing with oil spray injected from the nozzle O, became gasified in the vaporiser V and ignition tube T; the passage C communicated with the combustion chamber, and the main air inlet and exhaust valves were located as shown. B was the vapour valve, open during the suction stroke only; on compression the inflammable mixture was fired by the hot tube T heated by the blow-lamp L; no timing valve was employed. The whole apparatus was enclosed in a light metal outer casing. An early test by Messrs. Fielding & Platt, communicated to the author in 1894, furnished the results given in the following table:

TESTS OF A 3 HP NOMINAL FIELDING & PLATT OIL ENGINE MADE BY MESSRS. FIELDING & PLATT ON NOVEMBER 22 AND 23, 1894

Power	Full	Half	Light	Full
Duration of test	1 hour	1 hour	1 hour	3 hours
Revolutions per minute	220	225	230	222
Explosions per minute	100	84	18	100
Nett brake load	63 lbs.	33 lbs.	—	63 lbs.
Diameter brake circle	4 ft.	4 ft.	4 ft.	4 ft.
Brake HP	5·28	2·8	—	5·3
Oil per hour in lbs. (Russolene)	4·75	3·5	1·3	4·24
Oil per BHP hour	0·90 lb.	1·25 lbs.	—	0·80 lb.
Available pressure average of four diagrams, 79 lbs. per sq. in.				

Fig. 419 is a diagram from a similar 3 HP nominal engine taken in a test made on October 22, 1895, which also showed a consumption of 0·8 lb. of oil per BHP hour. During that test the engine gave 5·5 HP on the brake at 219 revolutions per minute, the compression was 40 lbs., and the maximum pressure of the explosion was 140 lbs., while the available pressure was 63 lbs. per sq. in. The oil engines of Messrs. Fielding & Platt are now (1911) fitted with automatic ignition, the blow-lamp being used for starting purposes only; by the courtesy of the makers the authors are enabled to show in complete detail the arrangement adopted.

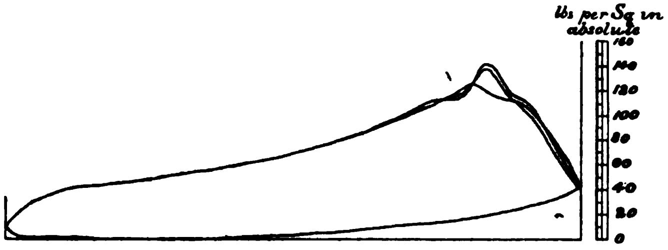


FIG. 419.—Fielding & Platt's Oil Engine (diagram)

Referring to fig. 420, the burnt gases passing the exhaust valve A are caused to circulate around the vaporiser B and ignition tube C before issuing at D; the explosion chamber X is thus exposed not only to the heat of the successive explosions, but is also jacketed with exhaust gas. The temperature of the vaporiser and explosion chamber may be regulated by aid of the throttle valve M, by which a less or greater proportion of the exhaust gas may be permitted to pass directly to the outlet D.

To start the engine the cap E is removed and the ignition tube and vaporiser are blow-lamp heated for about ten minutes as usual. G is an automatic spring-controlled air valve, and F is the fuel oil inlet; H is the cam-operated vapour valve (fig. 421). During the suction stroke of the piston H is open, and air then enters the vaporiser through the automatic valve G, taking with it the charge of oil sprayed

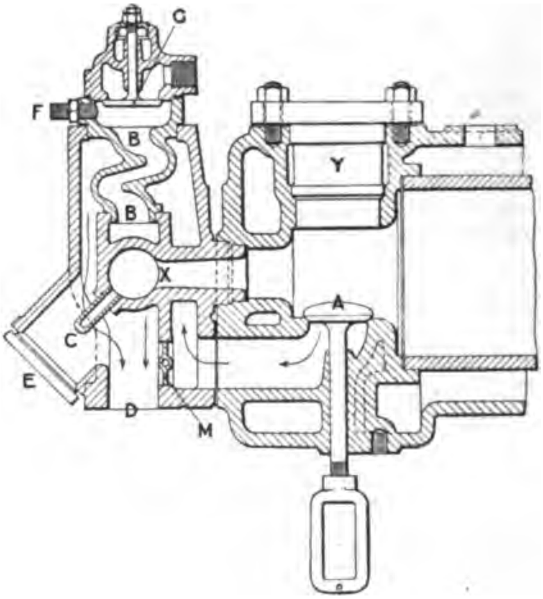


FIG. 420

in at the same time from F by the force-feed oil pump P, shown in fig. 421. The vapour valve is closed at the end of the suction stroke. The cam-operated main air valve K (fig. 422) is also open during the suction stroke. On the return of the piston the rich mixture in the explosion chamber X is compressed, and at the same time more and more diluted with the fresh air that entered through K; the proportions are so regulated that the mixture is automatically fired by the ignition tube C at the moment of maximum compression. The exhaust valve A is readily accessible for removal and grinding through the plug hole Y; the vaporiser is also easily removed for examination and cleaning.

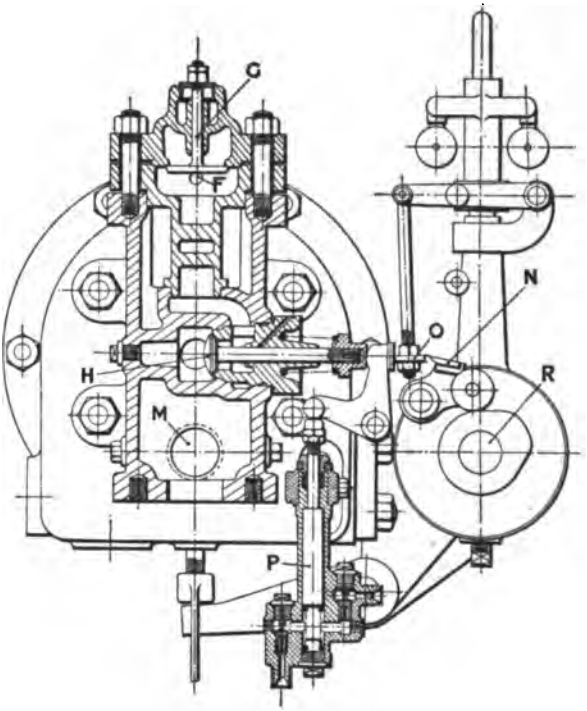


FIG. 421

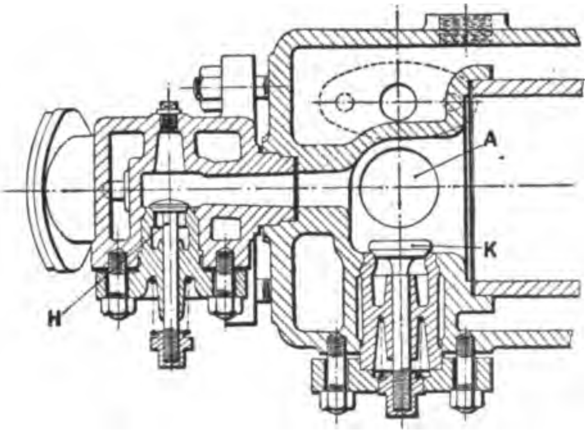


FIG. 422

The plunger of the fuel oil pump P (fig. 421) is actuated by the bell-crank as shown on the delivery stroke only, the return (suction) stroke being caused by a spring; standing alongside this pump, and operated by the same bell-crank lever, is a second similar pump, by which a small jet of water is injected into the cylinder through the main air valve K during the suction stroke, in order to soften the running and increase economy at full load.

Governing is on the hit-or-miss system; the arrangement of parts is clearly shown in the transverse sections, fig. 421; R is the half-speed shaft carrying a cam by which the bell-crank simultaneously operating the vapour valve, fuel oil pump, and water injection pump, is actuated through the steel knife-edge or 'pecker' N and grooved plate or 'pecker block' O. When the engine speed increases the governor moves the plate O so that it is missed by N, in which case the vapour valve remains closed and the oil and water pumps make no delivery stroke. The piston accordingly draws air only into the cylinder through the valve K on the suction stroke, and compresses this on the following stroke; no working stroke occurs, no oil is wasted, and by the arrangement adopted there is no risk of excess of water entering the cylinder; the disposition is simple, economical, and effective.

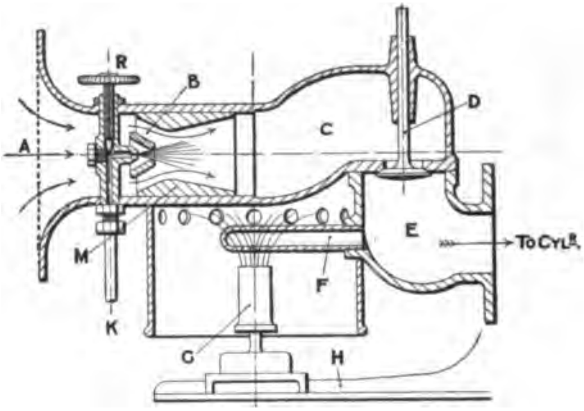


FIG. 423

The engine is built in the single-cylinder, horizontal type in sizes from 3 BHP at 400 r.p.m. to 60 BHP at 200 r.p.m., and with two cylinders to 120 BHP; as fuel, Russolene, Homelight, Solar oil, gas oil, shale oil, and ordinary prepared 'crude' oils may be used; the consumption at full load averages about 0·625 pint per BHP hour with refined oils.

The Robey Later Oil Engines.—In the earlier Robey oil engines the Norris automatic internal hot-bulb vaporiser and igniter already described was used. Later Messrs. Robey have employed a lamp-heated arrangement as illustrated diagrammatically in fig. 423. Air enters at A and, passing through the vena contracta B, induces a charge of oil delivered by the pump through a measuring device by way of the pipe K in the form of spray from the central nozzle as shown. The mixture passes into the chamber C, where it is vaporised, and thence viâ the vapour valve D—open during the suction stroke—to the explosion chamber E, which is in direct communication with the cylinder. Ignition is caused by the hot tube F, maintained in a state of incandescence by the blow-lamp G, which also constantly heats the vaporiser C.

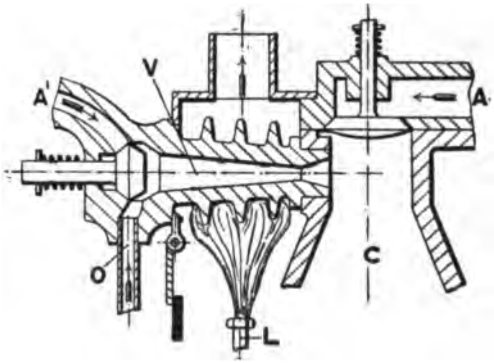


FIG. 424

The instant of ignition can be varied by shifting the lamp along the supporting rail H ; the quantity of air admitted to the vaporiser is regulated by moving the choke tube M to the right or left as required ; and the kerosene supply to the spraying nozzle is controlled by the regulating screw R.

In Messrs. Robey's latest practice the details of the device have been slightly modified, but lamp-heating is still retained for the vaporiser and ignition tube.

Governing is on the hit-or-miss system, the governor acting on the gear actuating the vapour valve, so that this does not open when the engine speed increases beyond the normal.

The Capitaine Oil Engines.—As already mentioned, the early Capitaine oil engines used internal hot-bulb vaporisers and igniters; in later practice this has been replaced by the simple form of lamp-heated vaporiser as illustrated in fig. 424. V is the vaporiser, heated by the lamp L. The oil pump delivery is at O, and a small quantity of air is admitted at A′; the main air inlet is at A; C is the combustion chamber.

The oil pump used in the Capitaine engine is of peculiar construction. Fig. 425 is a section. The plunger A is operated by bell-crank lever, roller, and cam, actuated in the usual way; and a slide valve B is actuated also by lever C and cam D; the plunger is packed by leather packing, and operates in a glycerine bath F. Oil G floats on the top of the glycerine bath, and is discharged through the slide valve B. In this way the plunger E is caused to operate in a space of ample capacity.

The Campbell Oil Engines.—In the early Campbell oil engines the lamp-heated vaporiser shown in fig. 426 was employed.

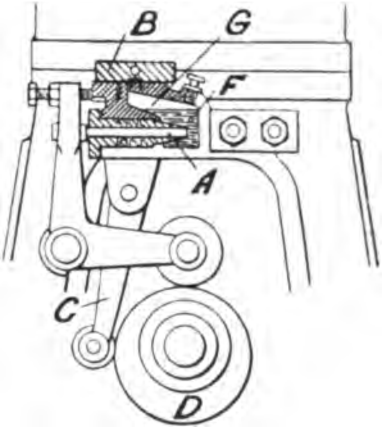


FIG. 425.—Capitaine Oil Engine (section through oil pump)

An automatic inlet valve A served for admission of the whole air charge to the cylinder by way of the vaporiser B and passage G. The oil was fed by gravity and passed through the supply pipe C to an annular channel D round the seat of the valve A, and was injected through perforations E to mix with the air when the valve opened. This valve, like the Tangye, resembled the gas and air valve first introduced by Clerk. On the suction stroke of the engine, air enters by the valve A, and oil entering with it is carried through the vaporiser, and the mixture of inflammable vapour and air passes into the engine cylinder by the passage G; this passage G leads into the exhaust port, as seen in fig. 427, and thus one port serves for the admission of the charge to the cylinder and the discharge of the exhaust products. The igniter tube H was screwed into the bend of the vaporiser and always in open communication with it. The lamp which heated the tube also heated the vaporiser, but while at work the heat of the explosions was sufficient to keep up the vaporiser temperature. The explosion ensues upon compression, the inflammable mixture being forced into the hot tube. This engine, however, was found to ignite without the tube after running for some time at full load.

The oil supply both to the vaporiser and to the lamp was by gravity, so that no oil pump was necessary; the engine had also but two valves, viz. the automatic inlet A (fig. 426) and cam-operated exhaust

J (fig. 427). A horizontal section is given in fig. 427, showing the exhaust valve and the exhaust port with the passage G from the vaporiser leading into the port. An engine of this type was tested at the Cambridge Royal Agricultural Show, and the full load results are given in the table on p. 690; the engine ran light at a speed of 211 revolutions per minute on an oil consumption of 2·32 lbs. per hour.

Fig. 428 is an indicator diagram taken from one of these Campbell engines during a two hours' test.

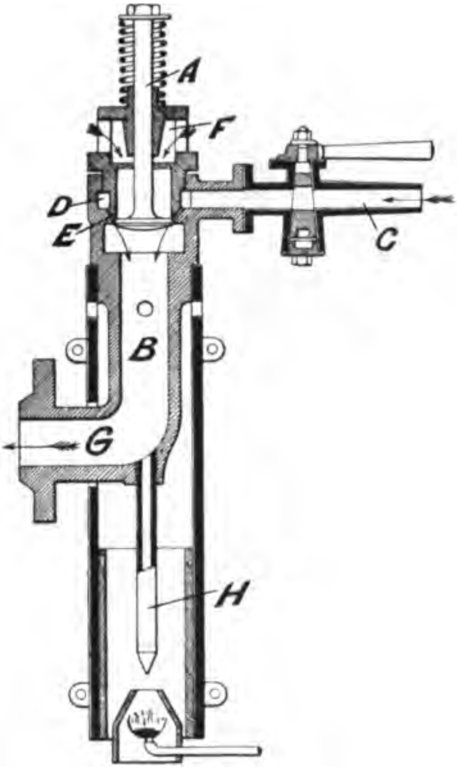


Fig. 426.—Campbell Oil Engine (section through vaporiser and igniter)

Governing was effected by holding up the exhaust valve so that no charge of oil and air was at such times sucked through the automatic valve A; the whole engine was of simple design. In their present (1911) oil engines Messrs. Campbell have modified the vaporiser and igniting tube just described in such manner that the ignition is now automatic, lamp-heating being only necessary for starting purposes. The automatic air inlet and gravity oil feed are still retained; the modification consists essentially in the replacement of the ordinary ignition tube H of fig. 426 by the thick-walled cast-iron tube shown in fig. 429, which, after being first lamp-heated for about ten minutes on starting, subsequently retains sufficient heat from the successive explosions to render the ignition automatic.

To diminish heat loss through radiation and provide also to some extent a means of regulating the temperature of this hot tube, a movable asbestos-lined sleeve is fitted over it. When the engine runs for prolonged periods at light load it is sometimes necessary to have recourse to the lamp. It will be noted that the igniting tube is drilled with two holes, that to the right being blanked at its lower end by a set screw, while that on the left communicates with an expanded portion in the lower part of the tube. The right-hand hole is considered to be that within which ignition originates, the function of the other, with the expanded lower portion, being to permit access of hot gas to the interior of the tube, and thus maintain it at a sufficiently high temperature to produce regular ignition.

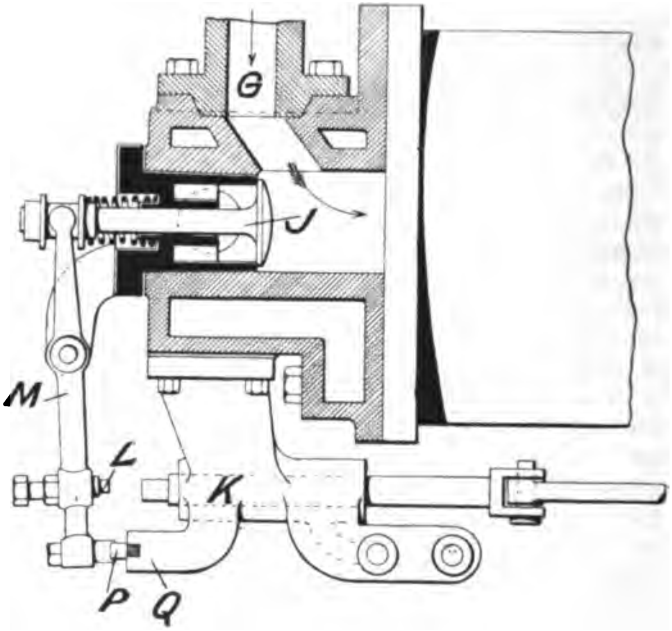


FIG. 427.—Campbell Oil Engine (horizontal section through exhaust valve)

A longitudinal section of the 9½ ins. × 16 ins. horizontal Campbell oil engine is shown in fig. 430, with the vaporiser in position. The water spraying valve will be noted; this is automatic in action and admits a small quantity of water from the cylinder jacket into the combustion chamber during the suction stroke; the quantity admitted is controlled by a small screw-down valve, and water injection is only used at full load.

The fuel oil supply is stored in a reservoir formed by the engine bed-plate as indicated, and the oil charge is drawn from this into the vaporiser by the piston suction alone through a supply pipe fitted with non-return valve and controlling cock.

Governing is on the hit-or-miss method (*v.* Chap. IV) and operates by holding the exhaust valve open, so that no charge is then induced on the suction stroke ;. the governor is driven from the camshaft by machine-cut spur gearing, while the camshaft is driven from the crankshaft by machine-cut helical gear. An external view of the normal 35 BHP Campbell oil engine is shown in fig. 431 ; this type of engine is built from 1½ BHP at 380 r.p.m. to 63 BHP at 180 r.p.m. ; and with two cylinders, side by side, to 126 BHP.

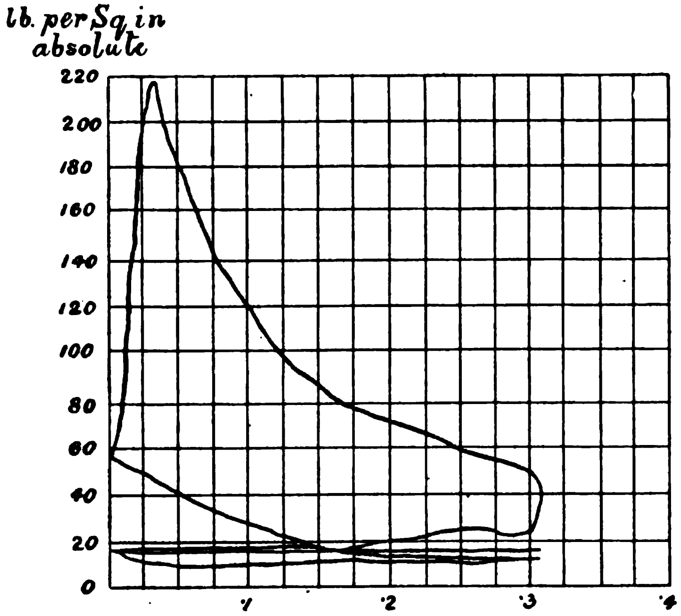


FIG. 428.—Campbell Oil Engine (diagram)

Type b_8*: The Diesel Engine.*—Of engines depending wholly upon the high temperature consequent upon very high compression for the vaporisation and inflammation of the working charge, that introduced by Herr Rudolph Diesel in 1893 and first described in his book entitled 'The Rational Heat Motor' (Eng. ed. by Spon) is typical.

The first Diesel engine, built by the Augsburg Co. in 1894, was of the inverted vertical, three-cylinder type ; among other early builders were Messrs. Carels Frères, of Ghent ; the engine was first publicly exhibited at the Munich Exhibition in 1898. It is now largely manufactured by many firms on the Continent, in America, and in Great Britain, among the latter being included the Mirrlees Diesel Co., Vickers (Barrow), John I. Thornycroft & Co., Willans & Robinson, and Richardsons, Westgarth & Co. One of the first Diesel engines to be built in England was of the horizontal two-stroke type, and was constructed by Messrs. Scott & Hodgson of Manchester for the Diesel Co. of England; this had a bore of 7⅞ ins. and stroke of 10¾ ins., and ran at 216 revolutions per minute, the corresponding piston speed being 387 ft. per minute only.

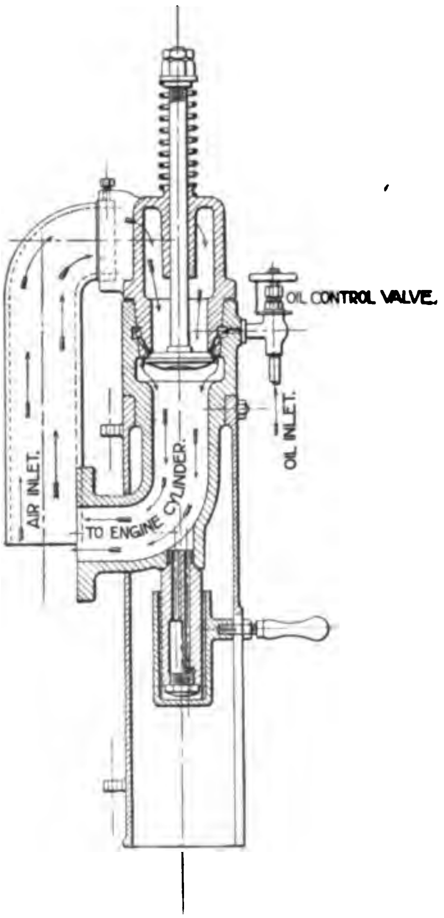
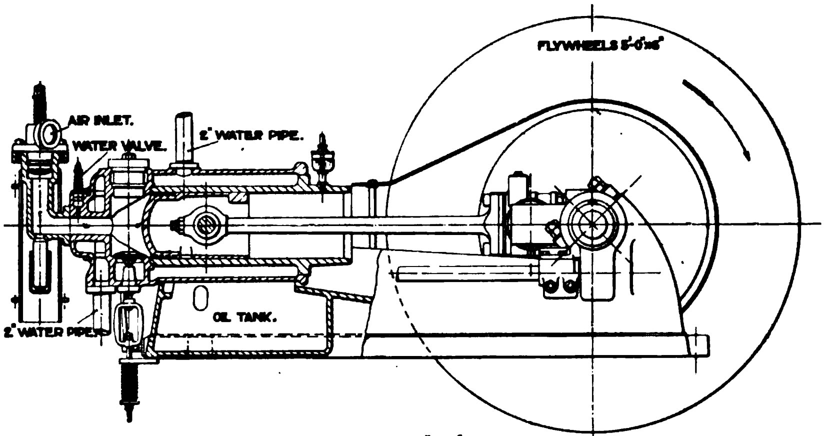


FIG. 429

On the piston rod, between the power piston and crank, was a second piston 9 ins. in diameter working in a cylinder; this piston during the first part of its in-stroke delivered air at a pressure of about 4 lbs. per sq. in. into a reservoir formed in the engine bed; this air was admitted to the power cylinder just after the overrunning of the exhaust ports by the power piston, and assisted in expelling the burnt gases.

This early two-stroke engine did not prove a success, but with increased experience in recent years Diesel engines of the inverted vertical type operating on the two-stroke cycle are becoming numerous and are likely to be largely used in the near future, especially in marine service; such engines have been greatly developed by Messrs. Sulzer, of Winterthur, within the past five years, and the Augsburg-Nurnberg Co. have also recently evolved a design of horizontal two- and four-cycle engines of about 400 BHP when worked on the Otto cycle. In marine applications the multi-cylindered vertical type operating on the two-stroke cycle is generally preferred, mainly on account of its easier reversibility and more uniform torque.

Great practical difficulties were at first encountered, arising from the very high pressures employed, from want of uniformity in the fuels used, from cracked and seized pistons, from imperfect and irregular combustion of the fuel, and from corrosion due to the presence of sulphur and other impurities in the fuel oils. The great pressure of the air necessary for the oil blast has also proved a fruitful source of trouble. Lubrication details and the preservation of gas-tightness in valves and pistons have also demanded much study. All these early difficulties have now been practically overcome, more particularly in four-cycle engines of up to about 200 HP per cylinder, as the result of extended experience and by the exercise of great skill in design and workmanship and careful selection of materials. The adoption of water-cooled stage air compressors has also done much to remove the difficulties of the air blast. Experiments have been conducted with a great variety of fuels, including petrol, kerosene, gas oil, crude Russian, American, and German oils; residual oils as Astatki, &c.; shale oils, coal tar oils, lignite oils, palm and nut oils, castor oil, fish oil, alcohol, coal gas, power gas, and coal dust. The greatest success has been attained with the kerosenes and heavier petroleum oils, and it is on these, and especially on the heavy dark-brown crude Texan fuel oil of sp. gr. about 0·925 and flash point 185° F., as largely used for firing steam boilers, that most Diesel engines in this country are now run.



LONGITUDINAL SECTION OF 9½"×18" CAMPBELL OIL ENGINE.

FIG. 430

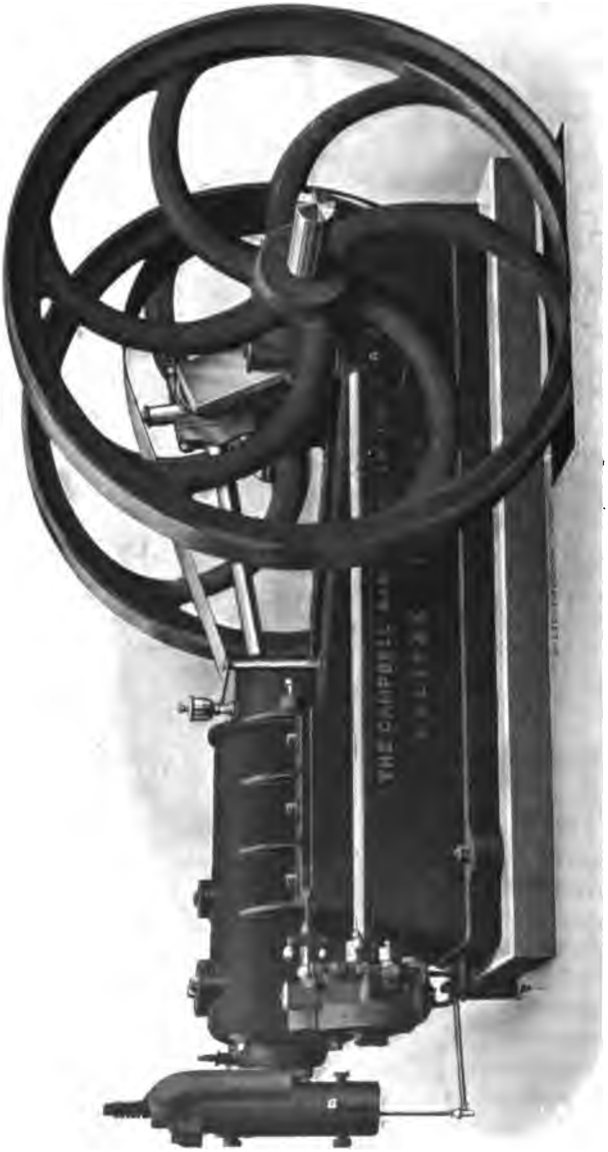


FIG. 431

Dr. W. Allner, of Dessau, in 1911, in a paper read before the German Gas Association, stated that the tar produced in large quantities in the gas and coke-oven industries has recently become practicable as a fuel for Diesel engines, the procedure consisting in injecting a small quantity of a readily ignitible auxiliary fuel, as Texan oil, either just before or simultaneously with the admission of the tar to the combustion chamber. The combustion of the auxiliary fuel starts that of the working fuel ; as auxiliary fuel gas oil is frequently used. In this way Dr. Allner states fluid crude tar may be utilised as fuel in Diesel engines. In general, however, it is a desideratum that the fuel employed be free from tar ; sulphur and acid impurities must also be absent. In the early coal-dust experiments a little liquid fuel was at first added in the hope of assisting combustion, but without practical success ; the engine ran, however, under experimental conditions, when the coal dust was mixed with a proportion of lean gas, and then furnished indicator diagrams of the normal type. Reproductions of diagrams obtained with various fuels are given on p. 595 of Vol. IV of the *Automotor Journal.*

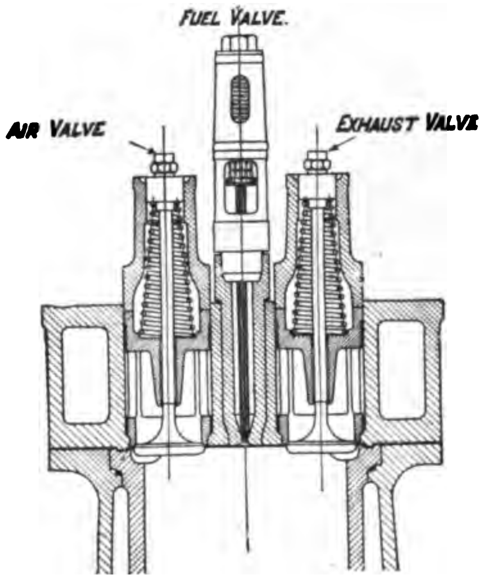


FIG. 432

It is a fundamental difficulty in all internal combustion engines in which the fuel, whether liquid or gaseous, is spray injected into the combustion chamber at or near the instant of maximum compression, to secure a uniform mixture with regular and complete combustion, and some of the earlier Diesel designs failed largely from this cause; the present practice is to locate the fuel-injection valve in the centre of the cylinder end, and so form the orifice—and in general the upper surface of the piston crown also—as to assist in causing an instant dispersion of the spray cloud uniformly around throughout the volume of the combustion chamber. Moreover, as this volume in Diesel engines must be very small on account of the very high compression pressure used, the air inlet and exhaust valves are also placed in the cylinder head. The normal arrangement is indicated

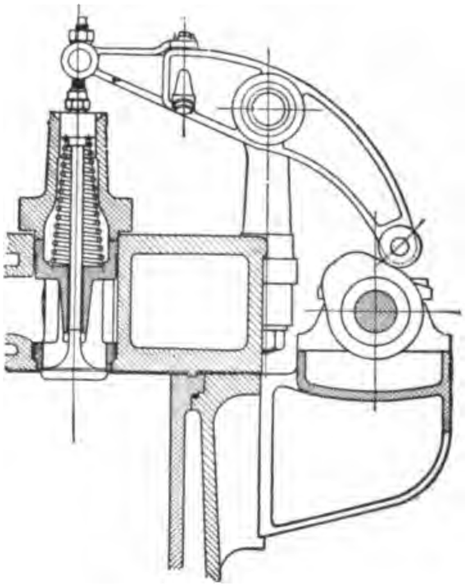


FIG. 433

in fig. 432, from which it will be noted that all three valves shown are contained in their own separate cages or castings, and can be readily removed, complete, from the cylinder head whenever necessary.

To operate these overhead valves the camshaft is located near the top end of the cylinders, a usual arrangement being as indicated in fig. 433. The bent rocking lever is constructed with a hinged upper portion which may be moved clear of the valve in the event of this latter requiring to be withdrawn for examination or adjustment.

Fig. 434 illustrates an early design by the American Diesel Co. which will serve for the purpose of description; the arrangement of valves shown did not prove practically successful; all valves are now placed in the cylinder head, as already mentioned. The air inlet valve, and exhaust valve E are inwardly-opening, cam-operated poppets of the usual type, while the needle valve P, through which the charge of atomised oil is sprayed into the combustion chamber by a blast of still more highly compressed air, opens outwards. It will be noted that the casing of P is water cooled to prevent vaporisation or carbonising of the fuel prior to its injection into the cylinder; this is a point of importance in working with some crude oils. Loosely surrounding the needle valve spindle is a series of perforated brass washers through and between which the liquid fuel percolates; these assist in the breaking up of the jet on entering the cylinder under the action of the air blast.

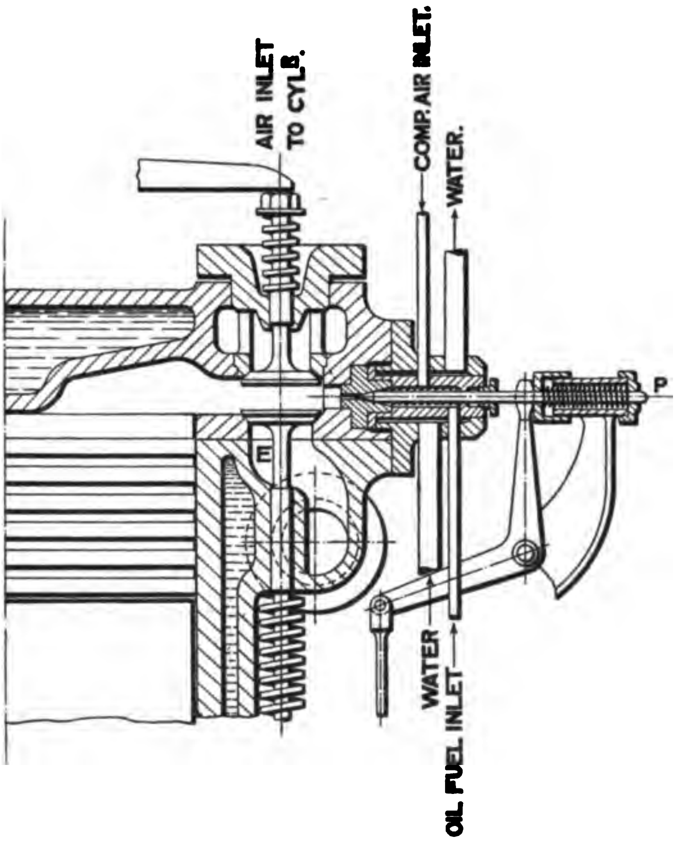


FIG. 434

An enlarged section of the usual design of fuel injection valve is

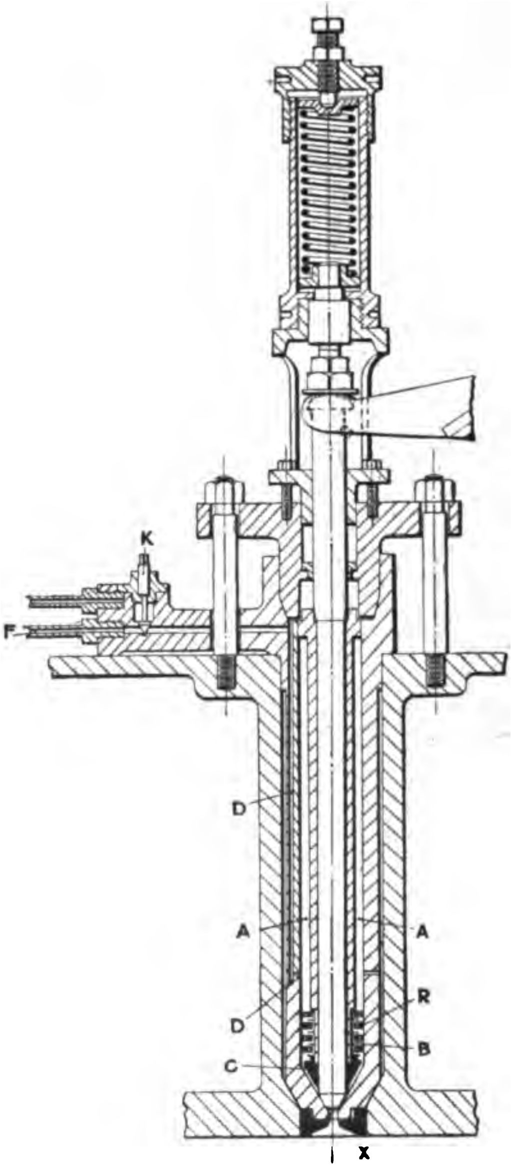


FIG. 435

given in fig. 435. The fuel force-pump delivers the charge of oil to the branch F, and thence along the duct D D into the annular space A A surrounding the cast-iron guide of the needle valve. The pulveriser comprises several brass washers R, each drilled with about twenty holes of one-sixteenth of an inch in diameter; these holes are staggered as indicated, so that the oil takes a tortuous path through them, and the rings are kept apart by distance washers B. Below these pulverising washers, and fitted to the bottom of the needle valve guide, is a conical head C, on the outer surface of which are formed some twenty channel grooves as indicated, through which the oil charge passes before reaching the spraying nozzle; immediately after passing the nozzle it enters the combustion chamber by the steel expanding orifice X.

The annular space A A is in constant communication with the air blast reservoir during the running of the engine, through a branch entering its upper portion which is not shown in the figure; thus this space is always under the high pressure of the air reservoir, and immediately the needle valve rises from its seat the charge of oil is blown with great velocity through the pulveriser washers and cone channels, and enters the combustion chamber through the expanding nozzle in the form of a uniformly diffused cloud of mist.

The small test cock K enables the attendant to ascertain that the oil supply is uninterrupted at any time.

On account of the serious pre-ignition effects that may result from leakage or sticking-up of the fuel inlet valve it is important that it be regularly and carefully cleaned at about fortnightly intervals; the frequency is, however, dependent upon the kind of fuel oil used. On account of the very small lift of this valve, and of the importance of admitting just exactly the right quantity of fuel at the right time, adjustments must be made with great nicety and skill.

The fuel commences to enter the combustion chamber just before the completion of the compression stroke, and at full load is injected during the first 20° or 30° of the crankshaft revolution; the fine spray cloud projected into the highly compressed and heated air in the combustion chamber instantly inflames, and the mixture burns at approximately constant pressure during admission. The engine speed is controlled by a governor actuating a by-pass valve in the fuel pump supply, whereby a larger or smaller quantity of the pumped oil is permitted to return into the suction pipe according as the engine is required to develop less or more power.

Herr Diesel originally proposed to carry out the Carnot cycle in his engine, the suggested order of operations being as follows:

(*a*) Isothermal compression of air using a water spray to preserve constancy of temperature.

(*b*) Adiabatic compression of the air from the lower to the upper limit of temperature.

(*c*) Regulated gradual injection of the fuel so as to produce isothermal combustion during the first portion of the working stroke.

(*d*) Further adiabatic expansion after fuel cut off, to the lower limit of temperature.

This would have furnished a diagram as given in Vol. I, fig. 30 (p. 93), with enormously high maximum pressure (Diesel contemplated pressures as high as 250 atmospheres) and very low mean pressure, and would have involved extremely large, heavy, and impracticable engines.

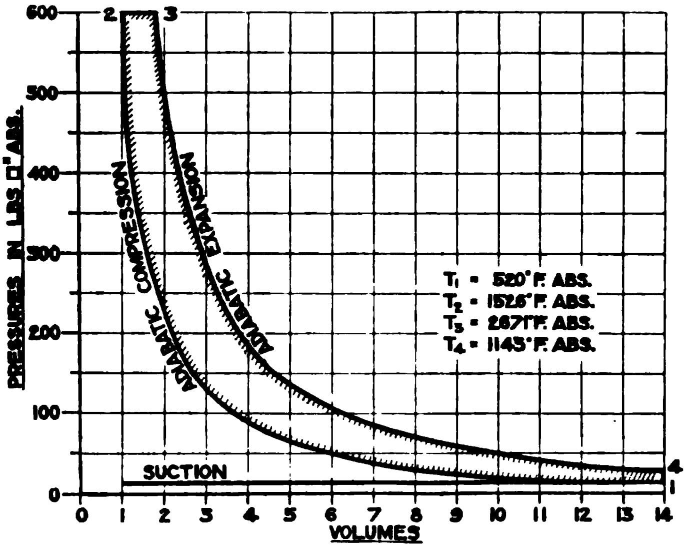


FIG. 436

Diesel also hoped that it would be possible to carry the expansion period (*d*) so far that no cylinder cooling water would be needed; but this was found impossible, and, in fact, the cylinder barrels and heads of these engines are in practice always exceedingly well water cooled. Practical difficulties also led to an early abandonment of the isothermal compression period (*a*) with water spray cooling and enormous compression pressure. In a paper read before the Congress at Paris in 1900, Diesel stated the cycle finally adopted after extended experiment as consisting of:

(*a*) A suction stroke (fig. 436) at atmospheric pressure.

(*b*) Adiabatic compression $\overline{12}$ to a pressure of 500–600 lbs. per sq. in.

(*c*) Regulated injection of fuel spray so as to maintain constant pressure combustion during the admission period $\overline{23}$.

(*d*) Expansion of adiabatic type $\overline{34}$ during the remainder of the working stroke.

(*e*) Exhaust $\overline{41}$ at constant volume, at the end of the stroke.

The air compressed is always considerably in excess of that necessary to burn the fuel injected in order to prevent the temperature during combustion from rising too high.

Fig. 436 shows, to scale, the diagram for such a cycle of operations; the volume ratio of compression is taken as 14, corresponding to a pressure after adiabatic compression from atmospheric, of about 600 lbs. per sq. in.; the heat admission period is shown as continuing during about 0·06 of the working stroke, and the expansion thereafter is assumed as adiabatic.

On these simplifying assumptions, heat $k_p\,(T_3 - T_2)$ is supplied during the admission period $\overline{23}$, and heat $k_v\,(T_4 - T_1)$ is rejected during exhaust $\overline{41}$, the air efficiency being consequently expressed by $k_p\,(T_3 - T_2) - k_v\,(T_4 - T_1)$ divided by $k_p\,(T_3 - T_2)$:

i.e. $$\text{Efficiency} = 1 - \frac{1}{\gamma} \cdot \frac{T_4 - T_1}{T_3 - T_2} \qquad (1)$$

The air being taken as initially at 60° F., the absolute temperatures are as given on fig. 436, and the efficiency in this case is therefore 0·613. The pressure at the end of expansion is only 32 lbs. per sq. in. absolute, and the mean effective pressure during the working stroke about 75 lbs. per sq. in., or one-eighth of the maximum pressure. The air standard efficiency corresponding to a volume compression ratio of 14 is 0·659; hence the efficiency of the cycle as assumed in fig. 436 is 93 per cent. of that of a perfect heat engine working between the limits T_1 and T_2. Observe from (1) that the efficiency is increased by diminishing T_4, i.e. by keeping the diagram as 'sharp-toed' as possible. It is of interest to note that the theoretic efficiency of this cycle can be expressed in terms of the adiabatic compression and constant pressure expansion ratios alone. For let $r = \frac{v_1}{v_2}$, and $\rho = \frac{v_3}{v_2}$ (Fig. 436); then it may easily be shown that:

$$T_2 = r^{\gamma-1}\,T_1\,; \qquad T_3 = \rho\, r^{\gamma-1}\,T_1\,; \qquad \text{and } T_4 = \rho^{\gamma}\,T_1.$$

On substituting these expressions for T_2, T_3, and T_4 respectively in Eq. (1) we obtain, after reduction:

$$\text{Efficiency} = 1 - \left(\frac{1}{r}\right)^{\gamma-1} \cdot \frac{\rho^{\gamma} - 1}{(\rho - 1)\,\gamma} \qquad (1')$$

which may be useful as enabling the efficiency to be calculated directly from the two leading ratios of the cycle, and independently of the temperatures.

As ρ tends to the value unity, i.e. as the constant-pressure expansion period is reduced, Eq. (1′) shows that the efficiency increases towards the limit $1 - \left(\frac{1}{r}\right)^{\gamma-1}$, which is the air standard efficiency corresponding to the adiabatic compression ratio $\frac{1}{r}$.

On the other hand, as ρ increases the efficiency diminishes from 0·659 when $\rho = 1$ to 0·253 when $\rho = r$, this latter value of ρ being of course quite impossible in practice. The frequently observed increase in the *indicated* thermal efficiency of the Diesel engine at reduced loads is in part due to the fact that with reduction in the constant pressure admission period the theoretical efficiency of the cycle increases, as just shown.

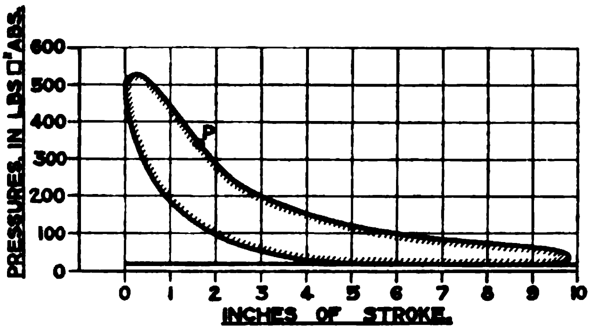


FIG. 437

In 1897 tests were conducted by Professor Schroter at Augsburg on an early vertical single-cylinder Diesel engine of 9·8 ins. bore and 15·7 ins. stroke, using as fuel an American kerosene of 0·796 sp. gr. The duration of each test was one hour; a reproduction of a full-load diagram is given in fig. 437. The compression pressure was roundly 500 lbs. per sq. in.; during admission the pressure rose slightly, a maximum of about 530 lbs. per sq. in. being attained; this was followed by a combustion period with falling pressure line until a point P was reached, the subsequent expansion being adiabatoid; Diesel, in 1900, regarded as desirable the maintenance of a constant pressure during the first part of admission followed by isothermal combustion during the latter portion, as approximately indicated in fig. 437. This diagram showed a mean effective pressure of about 110 lbs. per sq. in., i.e. about one-fifth of the maximum.

A summary of Professor Schröter's results is given in the following table, for reference:

TESTS OF A SINGLE-CYLINDER 9·8 INS. × 15·7 INS. DIESEL ENGINE, BY PROFESSOR SCHRÖTER IN 1897

(British Units)

Item	At			
	Full load		Half load	
Mean revs. per minute	171·8	154·2	154·1	158·0
MEP in cylinder, lbs. per sq. in.	106·0	105·0	75·0	73·0
IHP from cylinder	27·4	24·4	17·45	17·45
IHP from compressor pump	1·27	1·15	1·12	1·18
Nett IHP	26·13	23·25	16·33	16·27
Brake horse-power	19·6	17·55	9·44	9·7
Per cent. mech. eff.: $\frac{\text{BHP}}{\text{Nett IHP}}$	75·0	75·5	57·8	59·6
Lbs. of oil per IHP hour	0·396	0·383	0·336	0·343
Lbs. of oil per BHP hour	0·548	0·528	0·619	0·619
Cooling water inlet temp., ° F.	49·6	49·3	48·4	48·7
„ „ outlet temp., ° F.	78·8	68·4	65·7	68·1
„ „ rise in temp., ° F.	29·2	19·1	17·3	19·4
Lbs. of cooling water per IHP hour	103·5	157·2	171·0	121·0
Lbs. of cooling water per BHP hour	145·0	218·5	315·0	217·5
Exhaust gas temp., ° F.	759	712	500	500
HEAT BALANCE				
Per cent. appearing as IHP	35·3	36·5	41·5	40·7
„ appearing as BHP	*25·2*	*26·2*	*22·5*	*22·5*
„ appearing in cooling water	41·8	43·0	48·2	37·5
„ in exhaust gases, and loss	22·9	20·5	10·3	21·8
	100·0	100·0	100·0	100·0

The air compressing pump had a bore of 2·7 ins. and stroke of 7·8 ins., and diagrams taken showed a maximum delivery pressure of about 650 lbs. per sq. in.

The oil used had a percentage composition of 85·13 per cent. carbon and 14·21 per cent. hydrogen, and a calorific value (lower), by Junkers' calorimeter, of 18,250 BHP per lb.; to burn 1 lb. of this oil 14·7 8lbs. of air are theoretically necessary (*v.* Chap. IX); analyses of the exhaust gases furnished the following result:

	Per cent. by volume
N	85·14
CO_2	9·96
O	4·7
CO	0·2
	100·00

Thus the total volume of air supplied was $\frac{100}{79}$ N, while that necessary for combustion was $\frac{100}{79}$ N — $\frac{100}{21}$ O; hence the ratio of air supplied to air burned is expressed by:

$$\frac{\text{Air supplied}}{\text{Air burned}} = \frac{1}{1 - \frac{79}{21} \cdot \frac{O}{N}} \qquad (2)$$

and this had the value 1·26 at full load; at half load, the quantity of oil injected being reduced, the proportion rose to 2·16.

Trials of a single-cylinder, 30 HP Diesel engine, 11·8 ins. × 18·1 ins., made by Prof. Meyer in 1900 with (*a*) an American kerosene of 0·790 sp. gr. and (*b*) a Bavarian crude oil of 0·789 sp. gr., furnished the following results:

Prof. Meyer's Diesel Engine Trials, 1900

Item	American kerosene				Bavarian crude		
	Max. load	Normal	Three-quarter	Half	Normal	Three-quarter	Half
Revs. per minute	177·4	181·0	184·0	183·3	181·2	181·8	185·0
Indicated HP	47·5	38·9	32·6	24·7	40·3	32·5	26·0
Brake HP	38·9	29·7	23·5	15·0	29·9	23·2	15·2
Mechanical efficiency, per cent.	82·0	76·5	72·3	60·8	74·0	71·5	58·6
Lbs. oil per BHP hour	0·487	0·457	0·487	0·578	0·477	0·497	0·578
Brake thermal efficiency, per cent.	0·28	0·30	0·28	0·24	0·298	·28	0·24

Thus with this rather larger and later engine the brake thermal efficiency rises to 30 per cent. at normal full load, compared with 26 per cent. in the engine tested by Prof. Schröter.

One of the first Diesel engines to be installed for every-day work in England was the single-cylinder, 11·8 ins. × 18·1 ins., 35 horsepower, inverted vertical engine of the Harrogate Corporation, which commenced running in March 1902; in the following month tests of this were made by Mr. H. Ade Clark, of the Yorkshire College of Science, Leeds. As fuel a dark greenish-brown Texan crude petroleum was used, having a sp. gr. of 0·922 and calorific value 19,150 B.Th.U. per lb. Mr. Ade Clark made a second trial in 1903, after the engine had run for 16 hours daily during the intervening twelve months; some of his results are given hereunder:

FIRST TRIAL, APRIL 1902

Full Load

Revolutions per minute	181·4
Brake horse-power	40·26
Oil per BHP hour, lbs.	0·459
Brake thermal efficiency, per cent.	0·29
Duration of trial, minutes	40

The test was continued in May 1902, when indicator diagrams were taken, and a heat balance obtained; this furnished the following results:

Revolutions per minute	182·5
Indicated horse-power	52·27
Brake horse-power	39·21
Oil per IHP hour, lbs.	0·346
Oil per BHP hour, lbs.	0·461
Mechanical efficiency, per cent.	75
Duration of trial in minutes	60
Mean effective pressure, lbs. per sq. in.	117·6
Cooling water per IHP hour, lbs.	33·6
Rise of temperature of cooling water, ° F.	55·3
Heat Balance:	
Per cent. appearing as IHP	38·5
„ appearing as BHP	28·9
„ appearing in cooling water	28·1
„ appearing in exhaust gases	23·4
„ due to radiation losses, &c.	10·0
	100·0

SECOND TRIAL, TWELVE MONTHS LATER

At Nine-tenths Full Load

Revolutions per minute	179·9
Indicated horse-power	48·8
Brake horse-power	33·2
Oil per IHP, lbs.	0·344
Oil per BHP, lbs.	0·505
Mechanical efficiency, per cent.	68
Duration of trial, in minutes	50
Mean effective pressure, lbs. per sq. in.	106·5
Cooling water inlet temperature, ° F.	50
„ „ outlet temperature, ° F.	166
„ „ rise of temperature, ° F.	116

From the trial made in April 1902, Mr. Ade Clark has drawn

'Willans lines' for this 35 HP engine as shown in fig. 438; the total oil consumption between 15 and 40 BHP is proportional to the power, the oil per BHP being consequently given by the hyperbola B B.

A no-load test showed an oil consumption of rather over 7 lbs. per hour; hence the straight line A A curves round to A′ on the vertical axis somewhat as shown by the dotted line.

Tests were also conducted in 1903 by Mr. Ade Clark on a single cylinder, 15·75 ins. × 23·6 ins., 80 HP vertical Diesel engine at the works of the builders, Messrs. Carels Frères, of Ghent, and on a two-cylinder 160 HP engine of the same cylinder dimensions; in each of these cases the total oil per hour plotted against the BHP gave a straight line of the type as in fig. 438, the relation for the 80 HP engine being

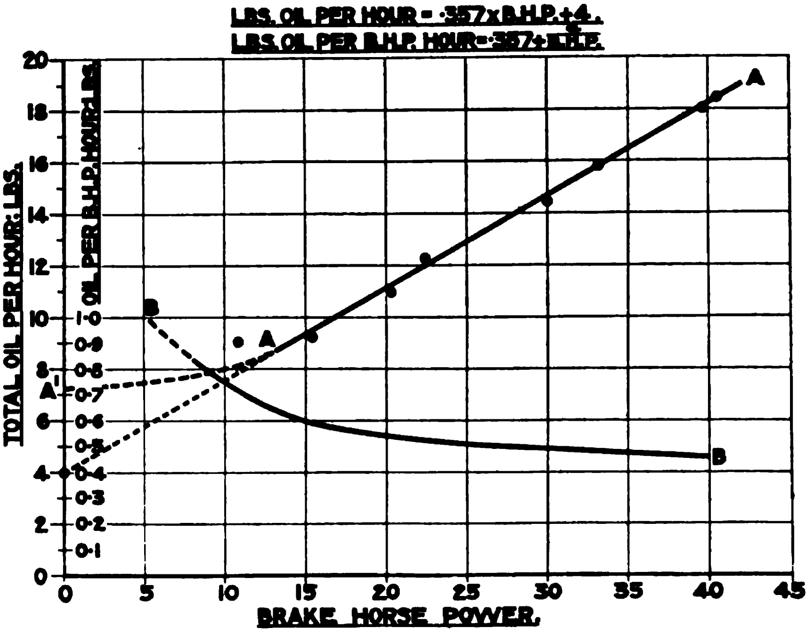


FIG. 438

$$\text{Lbs. oil per hour} = 0{\cdot}336 \times \text{BHP} + 7{\cdot}5 \tag{3}$$

while that for the 160 HP engine, both cylinders working, and for powers exceeding 50 HP was:

$$\text{Lbs. oil per hour} = 0{\cdot}345 \times \text{BHP} + 10 \tag{4}$$

The general dimensions of the single-cylinder, 80 BHP engines are given in the folowing table:

Item	Dimension
Cylinder diameter	15·75 ins.
Stroke of piston	23·62 ins.
Length of piston	35·65 ins.
Number of rings	7
Length of connecting-rod	63·4 ins.
Distance between crank-bearing centres	31·5 ins.
Distance between centres of flywheel bearings	56·7 ins.
Flywheel diameter	11ft. 2 ins.
Air-compressing pump, diameter	2·36 ins.
„ „ stroke	5·5 ins.
Oil air-blast reservoir (800 lbs. per sq. in.), diameter	8·0 ins.
„ „ „ length	35·4 ins.
Air reservoir for starting, diameter	13·4 ins.
„ „ „ „ length	70·3 ins.
Oil filter, diameter	13 ins.
„ „ length	23·6 ins.
Overall length of engine, complete	13ft. 2½ ins.
„ width „ „	13ft. 2½ ins.
„ height „ „	13ft. 2½ ins.

The engine was vertical with a stiff cast-iron A-frame whose upper portion formed the outer wall of the cylinder jacket; the working barrel fitted into this was of a special mixture of close-grained cast iron. The deep water jacketed cylinder cover contained four mechanically operated valves, viz. the needle oil injection, air starting, air suction inlet, and exhaust respectively; this disposition of valves is now general, enabling the very small clearance space necessary to be obtained; vertical valves also work better than horizontal. The constant-stroke plunger of the fuel oil pump was driven from the end of the camshaft; through the head of the plunger passed a lever rocking about a fulcrum situated eccentrically upon a shaft connected with a loaded centrifugal governor; to this lever was attached a rod which opened the pump suction valve against the action of a closing spring; by shifting the fulcrum the governor thus varied the degree of lift of the suction valve and hence the quantity of oil admitted to the pump chamber during the suction stroke.

The air-compressing pump was water jacketed and its plunger was driven by link-work from a point near the top of the connecting-rod; this pump took its air supply from the engine cylinder just before the end of the compression stroke, still further compressed the air, and delivered it to the oil air-blast reservoir at a pressure about 150 lbs. per sq. in. in excess of the maximum cylinder compression pressure; by this means the pump was kept of small size. The practice of taking partly compressed air from the cylinder by the compressor pump was not practically satisfactory, and is not now employed. The engine is started by compressed air stored in the starting reservoir by the air-pump during the preceding running of the engine to a pressure of about 800 lbs. per sq. in.; in the case of a new engine the starting reservoirs are sent out fully charged by the makers; the loss of pressure is very small, an engine sent out to India, for example, being started from a reservoir charged in Belgium four months previously. By means of a hand lever the engine is racked over until the crank-pin is just over the top centre; by means of a starting lever the starting cam is next brought into working position; on opening the blast reservoir and starting reservoir valves the engine at once moves off under the air pressure; after a few revolutions the starting lever is moved back, and the engine continues to run on its normal working cycle. The full revolution rate is attained at once, but the normal power output is not reached until the engine is well warmed up. In fig. 439 is shown an indicator diagram from the two-cylinder, 160 HP engine showing four air-pressure strokes followed by the first three firing strokes; the maximum air pressure in the cylinder during the first stroke was about 475 lbs. per sq. in., falling to about 325 lbs. per sq. in. at the fourth stroke.

A trial of the single-cylinder, 15·75 ins. × 23·6 ins., 80 HP Diesel engine at Ghent by Mr. Ade Clark in 1903, using as fuel a Texan crude oil of 0·922 sp. gr. and calorific value 19,300 B.Th.U. per lb., furnished results as under:

SINGLE-CYLINDER, 80 HP DIESEL ENGINE TRIAL AT GHENT. (*Ade Clark*)

Item	At load:			
	¼	½	¾	Full
Duration of trial, minutes	30·5	30·1	32·0	30·75
Mean revs. per minute	163	161	160	160
Mean effective pressure, lbs. per sq. in.	48·5	68·4	89·1	109·4
Indicated horse-power	46·8	67·0	82·8	101·5
Effective HP = (IHP − 22)	24·8	45·0	62·8	79·5
Mechanical efficiency. per cent.	53·0	67·0	73·5	78·3
Lbs. of oil per IHP hour	0·352	0·320	—	0·339
Lbs. of oil per effective HP hour	0·664	0·477	—	0·434
Indicated thermal efficiency, per cent.	37·5	41·2	—	38·9
Effective thermal efficiency, per cent.	19·9	27·7	—	30·4

A month later Mr. Ade Clark also tested a two-cylinder, 15·75 ins. × 23·6 ins., 160 HP engine, using the same fuel, at Ghent, with results as follows:

TWO-CYLINDER, 160 HP DIESEL ENGINE BY CARELS FRÈRES, GHENT

Item	At load				
	0	¼	½	¾	Full
Duration of trial, minutes . .	30	60	61	60	60
Mean revs. per minute . . .	159·0	158·0	158·0	157·0	154·5
Mean eff. pressure, lbs. per sq. in. (av.)	43·0	42·1	56·3	69·4	114·0
Indicated horse-power, total . .	39·64	74·9	100·0	126·8	204·4
IHP of air comp. pump . . .	2·98	2·97	3·0	3.25	3·3
Nett IHP	36·66	71·93	97·0	123·6	201·1
Effective HP (total IHP – 39·6) . .	—	35·3	60·4	87·2	164·8
Mechanical efficiency, per cent . .	—	47·2	60·4	68·8	80·7
Oil per nett IHP hour . . .	0·415	0·356	0·309	0·329	0·333
Oil per effective HP hour . .	—	0·725	0·496	0·467	0·406
Rise of temp. of cooling water, ° F. .	102·6	100·8	91·8	77·4	94·5
Exhaust gas temperature, ° F. . .	—	316·0	387·0	462·0	723·0
Nett indicated thermal eff., per cent.	31·8	37·1	42·7	40·2	39·7
Effective thermal efficiency, per cent. .	—	18·3	26·6	28·3	32·6

The outlet temperature of the cooling water at full load was 153° F. Analysis of the exhaust showed no perceptible CO at any time; at full load the percentage volumes were: N = 81·7; O = 11·3; CO_2 = 7·0, corresponding to a ratio of air supplied to air burned (Eq. (2), p. 726) of 2·09; the exhaust was at all times clear and free from smell of unburnt oil; Mr. Ade Clark takes the specific heat of the exhaust gases at constant pressure as of the value 0·25 B.Th.U. per

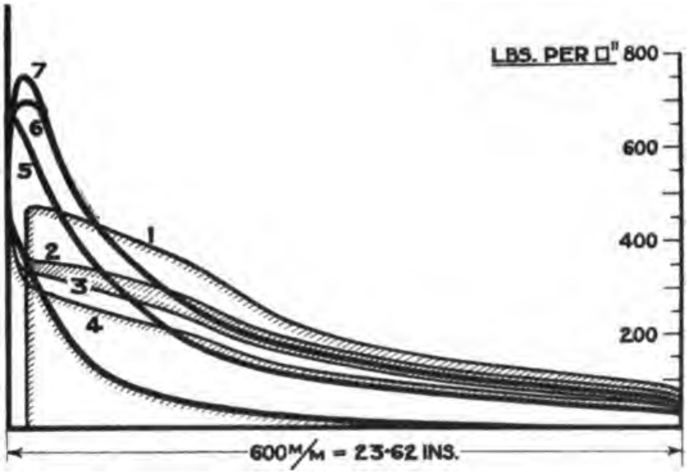


FIG. 439

° F. per lb. In the no load trial only one cylinder was power producing, the other pumping idly. Messrs. Carels Frères have used the Reavell four-cylinder, three-stage air compressor largely in connection with their Diesel engines; these compressors will deliver air at pressures up to 1000 lbs. per sq. in., with a volumetric efficiency approaching 90 per cent., and a high mechanical efficiency.

In February 1905 Mr. M. Longridge, M.Inst.C.E., conducted tests on a 22·05 ins. × 29·52 ins. three-cylinder, vertical, 500 HP Diesel engine built by Messrs. Carels Frères, which was subsequently on view at the Liège Exhibition. Some of his results are given in Vol. I of this work (p. 320), where the author has pointed out an error in procedure very commonly made in estimating the indicated thermal efficiency and the mechanical efficiency of the Diesel engine. If the blast air pressure were in some way produced in the power cylinder, the necessary work would appear in the compression curve of the power cylinder indicator diagram by raising this curve, and thus diminishing the area of the positive part of the diagram, and consequently also the IHP as estimated therefrom.

The fact that the blast air pressure is not so obtained does not alter the matter, and the conclusion is that the work shown by the blast air pump diagram must be deducted from that shown by the power cylinder diagram in estimating the IHP.

This has the effect of reducing the figure expressing the indicated thermal efficiency and—when the blast pump is driven by the engine—increasing the figure for mechanical efficiency, the brake thermal efficiency remaining unaltered. Corrected in this way, Mr. Longridge's tests lead to the conclusion that at full load the IHP of the engine was 595; BHP 459; mechanical efficiency, 77 per cent.; indicated thermal efficiency 41 per cent.; and brake thermal efficiency 31·7 per cent. The maximum pressure shown by the indicator diagrams was 525 lbs. per sq. in., and the greatest mean effective pressure obtained was about 115 lbs. per sq. in.; the ratio of this maximum to mean is thus roundly 4½ : 1. The blast pressure attained the very high value of 975 lbs. per sq. in. A Galician petroleum was used as fuel.

The crankshaft was 11 ins. in diameter, carried in four white-metal lined bearings, ring-lubricated. The connecting-rods were of marine type with gudgeon pins 7·9 ins. dia. × 12·6 ins. long in a phosphor bronze bearing, and crank pins 11·8 ins. dia. × 12·6 ins. long in white metal bearings. Each piston was fitted with eight spring rings, the eighth being low down, and was lubricated by oil forced through a ring of small holes in the lower part of the cylinder barrel. The air suction, exhausts, oil needle, and air starting valves were all located in the cylinder head; the air suction and exhaust valves were self-contained in cages, valve and seat thus being readily removable and replaceable; a harder and closer metal can also be obtained for the valve seats thus separately cast; the exhaust valves were water cooled. The lever actuating the oil needle valve was in two parts, allowing ready examination of the valve to be made without disturbing the camshaft. The normal full speed was 150 revolutions per minute, corresponding to a piston speed of roundly 740 ft. per minute.

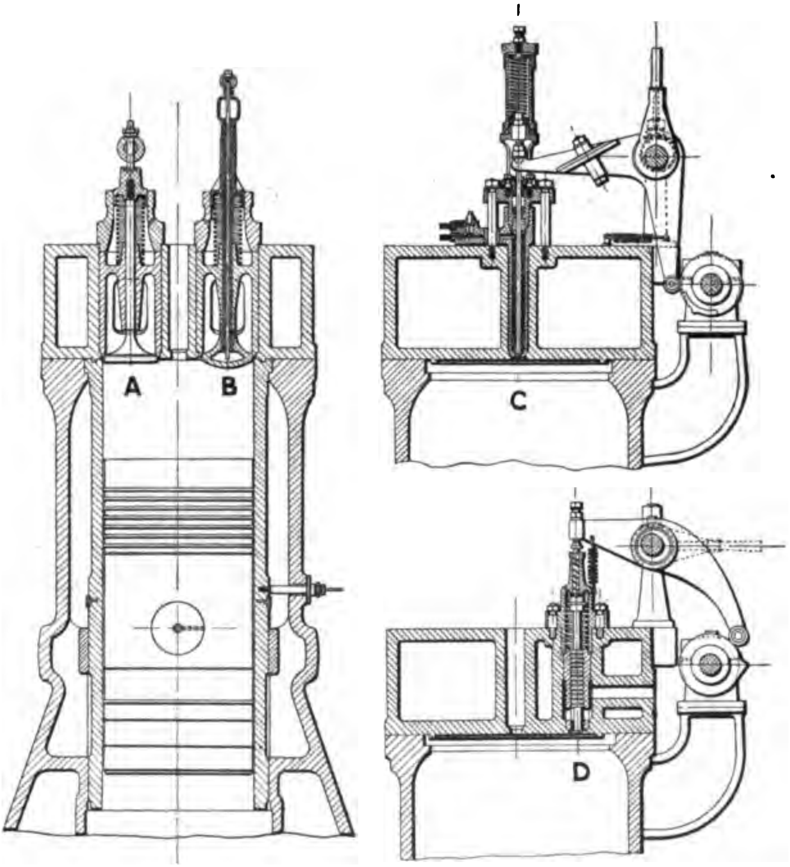


FIG. 440

Sectional views of the air suction, water-cooled exhaust, oil needle, and air starting valves are shown in fig. 440 at A, B, C, and D respectively; A shows also the general arrangement of the cylinder, head, and casing; the two-part lever actuating the oil needle valve will be noted.

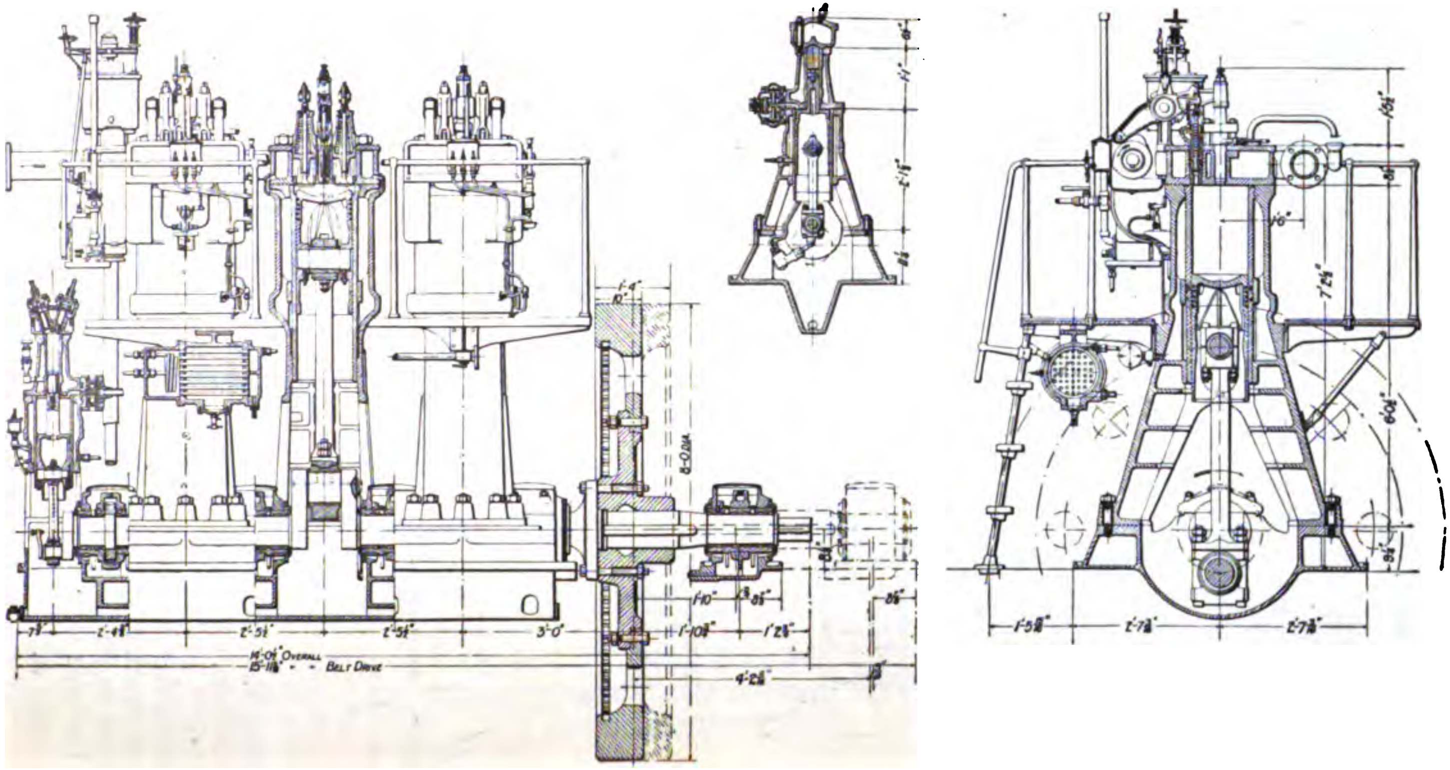

7⅛"
2'-4⅛"
2'-5½"
2'-5½"
3'-0"
14'-0¼" Overall
15'-11⅛" " " Belt Drive
1'-4"
10"
8'-0 DIA.
1'-10"
8⅝"
1'-10⅜"
1'-2⅝"
8⅝"
4'-2⅝"
8½"
1'1"
2'-1⅝"
8⅛"
1'5½"
8⅝"
1'6"
7'-2¼"
6'-0½"
9½"
1'-5⅝"
2'-7⅜"
2'-7⅜"

FIG. 441

The Mirrlees Diesel Engines.—Among the earliest British builders of Diesel engines were Messrs. Mirrlees, Watson, & Co., Ltd., of Glasgow; the Mirrlees engines are now constructed by Messrs. Mirrlees, Bickerton, & Day, Ltd., Stockport. The standard open, vertical, three-cylinder, 12 ins. × 18½ ins. Mirrlees-Diesel engine, developing 120 BHP at its normal speed of 200 revolutions per minute, is illustrated in the accompanying plate, fig. 441.

Very stout cast-iron A-frames support the cylinders, the upper part of the frames forming the outer jacket walls; the jackets extend below the end of the piston stroke; the deep cast-iron water-cooled separate cylinder heads contain each the four valves carried in separate cast-iron cages as usual.

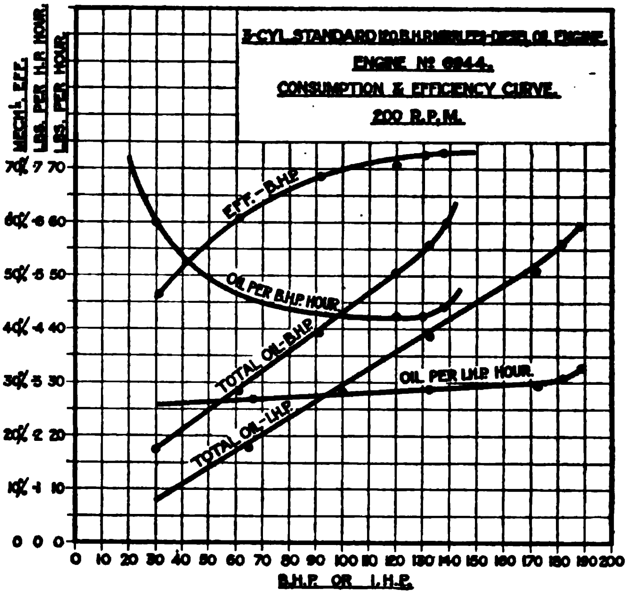


FIG. 442

The pistons are of cast iron, 26½ ins. in length, each with five spring rings, the dished crowns being supported by webs extending to the gudgeon bosses; the oil is injected vertically downwards from the centre of the cylinder head, and a small domed eminence will be noted on the piston crown immediately beneath; the general thickness of the crown is 1¼ ins. The gudgeon pins are of steel, 4⅛ ins. dia. and 6½ ins. long, with bronze bearings; the connecting-rods are of steel, of normal marine type, with a mean shank diameter of 3¾ ins.; the length of the connecting-rods is 4 ft.; the crank-pins are 6¼ ins. dia. × 6¾ ins. long;

big ends cast iron, white metal lined. The crankshaft is 6¼ ins. dia., borne in four cast-iron white metal lined bearings; a fifth similar bearing supports the shaft beyond the 8 ft. × 10 ins. cast-iron flywheel; when the engine is used for driving electric generators the flywheel rim is increased to 16 ins. in width to obtain the necessary uniformity of motion; the diameter of the crankshaft in the flywheel boss is 7¾ ins. The main bearings are ring lubricated, two rings being fitted in each; the big ends are centrifugally lubricated from a light annular channelled casting attached to the crank-cheek and communicating with the bearing surfaces through drilled ducts in the crank-pin, as indicated in fig. 441. The piston lubrication is by the usual ring of holes in the lower portion of the working barrel, supplied with oil under pressure by a small lubricating pump.

The fuel oil pumps, one to each cylinder, are driven from the half-speed shaft near the top of the engine; the supply of oil is governor-regulated, the pump suction valve being held off its seat during a portion of the delivery stroke varying with the speed of the engine. The two-stage cast-iron, water-cooled air compressor, shown in section on the extreme left of fig. 441, is driven from the end of the crankshaft, and not by link-work from the connecting-rod as in the earlier designs; after the first compression the air is cooled, with separation of contained moisture; inter-stage cooling increases the efficiency of the pump, and by the reduction of the temperature improves the working of the valves on the high pressure pump cylinder. The air compressor delivers into storage reservoirs, whence is drawn the 'air blast' for injecting the charge of oil into the engine cylinders. The stored air is used also for starting purposes, as already mentioned.

The approximate extreme dimensions of this engine are: Length 14 ft. 3 ins.; width 8 ft. 3 ins.; and height above ground level 10 ft. 0 in.

It is adjusted, when circumstances require, to also run at a normal speed of 250 revolutions per minute, the BHP being then 150, or 50 horse-power per cylinder; units of 1, 2, 3, and 4 such cylinders respectively are standard Mirrlees-Diesel arrangements.

Consumption and efficiency curves for the three-cylinder, 120 HP engine of fig. 441 at 200 revolutions per minute are shown in fig. 442; the graph was kindly supplied to the authors by the Mirrlees Diesel Company.

It will be noted that the total oil per hour plots against the BHP sensibly as a straight line from 30 HP to 130 HP, and over this range we have:

$$\text{Lbs. oil per hour} = 0{\cdot}362 \times \text{BHP} + 7 \qquad (5)$$

The mechanical efficiency at 130 BHP is about 73 per cent; at this power also the oil consumption is 0·425 lb. per BHP hour; taking the calorific value of the oil at roundly 19,000 B.Th.U. per lb., this gives a brake thermal efficiency of 31·6 per cent.

In fig. 443 indicator diagrams at full, three-quarter, half, and quarter loads are shown from an engine of this type running at 250 revolutions per minute, the output at full load being then roundly 150 BHP. The power was absorbed electrically, the dynamo efficiency being 90·3 per cent. Assuming the diagram shown for full load to apply to each of the three cylinders, the mechanical efficiency comes out at 74 per cent., which agrees well with the efficiency curve shown on fig. 442.

The authors are indebted to the Diesel Engine Company, Ltd., of London for the following test results obtained in October 1910 from a three-cylinder, four-cycle Diesel engine of 17·75 ins. bore and 27 ins. stroke installed in the electric power house of the G.N. Railway Co. at Doncaster. The engine is coupled direct to a 240 kilowatt dynamo, 220 volts.

TEST OF THREE-CYLINDER, 360 BHP DIESEL AT G.N. RAILWAY WORKS, DONCASTER, 1910

Item	At quarter load	At half load	At three-quarter load	At full load	At 10 per cent. over-load
Date of trial . . .	25/10/10	25/10/10	25/10/10	24/10/10	25/10/10
Duration in hours . .	1	1	1	6	1
Revs. per minute . .	190	190	190	190	190
Piston speed, ft. per min. .	855·0	855·0	855·0	855·0	855·0
Indicated horse-power .	214	301	387·0	485	541·0
Brake horse-power . .	—	178·7	262·2	346·0	380·0
Mechl. eff., BHP/IHP .	—	59·3	67·9	71·3	70·3
Total fuel oil used, lbs. .	61·25	88·5	119·0	921·0	177·8
Lbs. fuel per IHP hour .	0·286	0·294	0·308	0·317	0·329
Lbs. fuel per BHP hour .	—	0·495	0·454	0·443	0·468
Average MEP, lbs. per sq. in.	44·6	62·7	80·7	101·0	112·8
Mean blast pressure, lbs. per sq. in.	718	746	782	863	917
Total cooling water, lbs. .	7450	9850	11,100	115,650	15,800
Lbs. water per BHP hour	—	55·0	42·3	55·7	41·6
Inlet temp. of water, ° F. .	—	—	—	54	—
Outlet temp. of water, ° F. .	—	—	—	103	—
Rise of temperature, ° F. .	—	—	—	49	—

At full load the cost of the fuel oil per BHP hour amounted to only 0·108 pence; and per electrical unit generated 0·156 pence. It will be noted also that the indicated efficiency diminishes steadily with increase of output from one-quarter load to 10 per cent. overload. At the end of the six hours' full load run the governor was tested by suddenly throwing the load off the engine; there was a momentary rise in speed from 190 to 204 revolutions, being a 7·4 per cent. increase; the engine settled down without load to a speed of 195 r.p.m., corresponding to a 2·63 per cent. increase only.

Reproductions of indicator diagrams taken during these trials are given in fig. 444.

In a paper read before the Institution of Electrical Engineers in May 1909, Mr. A. J. Pfeiffer stated that there was then in service a vertical, four-cylindered, 800 BHP, single-acting, four-cycle Diesel engine running at 150 revolutions per minute, and that this was probably about the limit of size which can be satisfactorily run without water- or oil-cooled pistons and exhaust valves. At the Daimler Motor Company's works at Coventry two 300 BHP Diesel engines have for some years past been regularly employed in driving 50-cycle alternating generators in parallel.

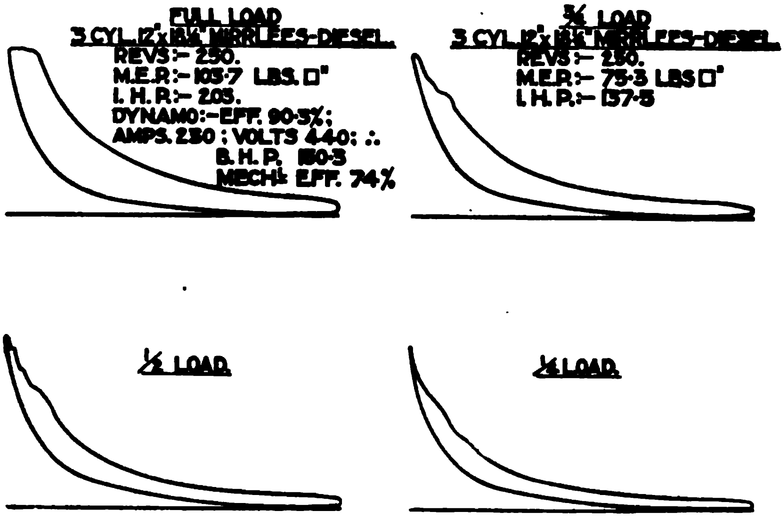


FIG. 443

Weight of Diesel Engines.—Although the maximum pressure and greatest mean effective pressure (about 550 and 115 lbs. per sq. in. respectively) which occur in Diesel engines in normal conditions of working are only, at most, about 50 per cent. greater than are found in many gas and petrol engines, these engines are yet usually characterised by their exceedingly massive construction.

Considerations of economical production have caused cast iron to be very extensively employed, and with this material, weak in tension, heavy scantlings become necessary. Pre-ignition occasionally occurs also, due sometimes to leakage of oil past the needle valve, or to this valve sticking up, and sometimes in two-stroke engines after a period of light load running with, e.g. two of the four cylinders not firing. On first again injecting the oil into these idle cylinders an explosion may be missed, and the next compression of the thus formed inflammable mixture results in a pre-ignition, the pressure produced being further augmented by the normal oil injection when the top of the stroke is reached. In this manner momentary pressures may be produced in the cylinder of 1000 lbs. per sq. in. or more, and this has to be kept in mind in design.

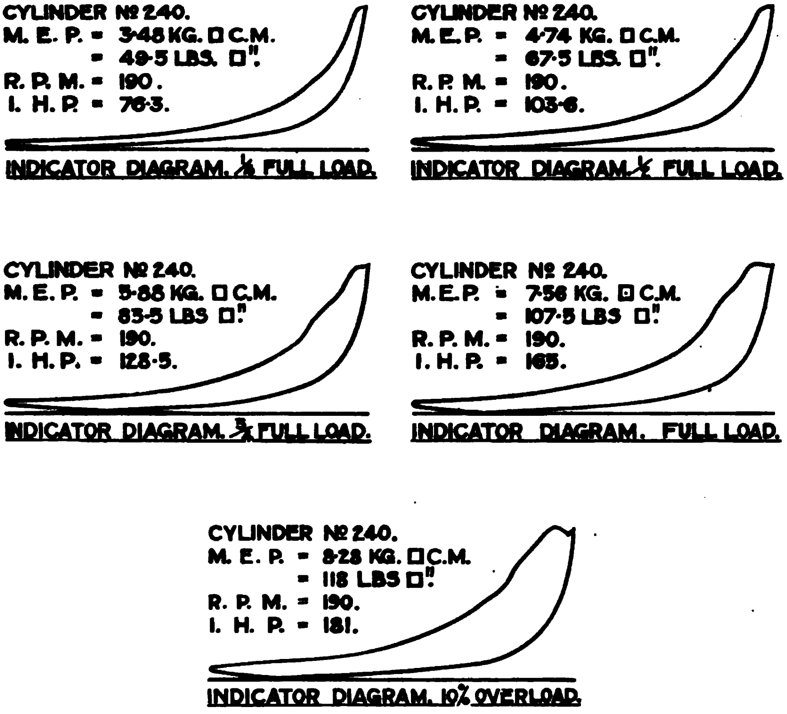


FIG. 444

A sticky needle valve, moreover, enables the very high pressure blast air to enter the cylinder in abnormal quantities, and this may also cause failure of the cylinder through extreme pressure caused by its subsequent further compression by the rising piston; Mr. D. M. Shannon[1] states that within his experience at least two engines have recently been wrecked, apparently from this cause alone.

Due, again, to the high compression employed very heavy flywheels become necessary, especially with few-cylindered, four-cycle engines, in order to reduce the cyclic irregularity (see Chap. IV); with multi-cylindered two-stroke engines the flywheel may be much reduced, especially when such engines are made double acting; the elimination of the flywheel is a desideratum in marine service. The following figures, from Guldner, enable usual practice in the matter of engine weight to be appreciated; the figures refer to standard engines designed for land installations.

---

[1] *Gas and Oil Power*, June 6, 1912.

Total Weight of Augsburg-Nürnberg Diesel Land Engines, including Flywheel

(1) *Single-cylinder engines*

BHP	4	10	20	30	40	50	60	70	80	100
Total wt., lbs.	2420	4620	11,450	17,200	23,300	28,600	35,200	41,800	47,200	59,400
Lbs. per BHP	605	462	572	574	583	573	587	597	590	594

(2) *Double-cylinder engines*

BHP	30	50	70	100	140	200
Total wt., lbs.	15,400	25,300	35,800	51,600	74 800	105,500
Lbs. per BHP	513	507	512	516	535	527

Weight of American Diesel Co.'s Engines with and without Flywheels. Inverted Vertical Land Type

No of cylinders	1	3	3	3	3
Brake HP	75	75	120	170	225
Bore of cylinders	16	10½	12	14	16
Stroke of piston	24	15	18	21	24
Revs. per min.	164	240	220	200	164
Weight without flywheels	28,600	18,500	27,000	55,000	65,000
No. of flywheels	1	2	2	2	1
Total flywheel weight	11,800	5400	11,800	13,800	15,640
Total weight of engine	40,400	23,900	38,800	68,800	80,640
Engine weight, lbs. per BHP without flywheels	382	247	225	324	289
Engine weight, lbs. per BHP with flywheels	540	319	323	405	358

Thus for land Diesel engines up to 200 BHP the weight per BHP is roundly, including flywheels:

For single-cylinder engines	600 lbs.
For two-cylinder engines	520 lbs.
For three-cylinder engines	350 lbs.
For three-cylinder engines without flywheels	275 lbs.

For comparison, a short table is given hereunder showing the weight per normal BHP of a number of well-known stationary oil engines of usual type; the weights given include flywheels:

WEIGHTS OF OIL ENGINES, INCLUDING FLYWHEELS

Engine	No. of cyls.	Normal BHP	Revs. per minute	Approx. weight	
				Total, in cwts.	Lbs. per BHP
Crossley, ordinary type	1	5	330	13	290
Petter	1	10	250	32	360
Fielding	1	30	220	74	275
Petter	1	38	225	113	335
Robey	1	40	210	111	310
National, ordinary type	1	40	230	104	290
Tangye, crude oil	1	44	200	118	300
Crossley 'industrial'	1	47	200	145	345
Campbell, ordinary type	1	54	200	160	335
Fielding	1	60	200	207	385
Ruston, ordinary type	1	60	190	188	350
Campbell, ordinary type	2	126	180	330	295

Broadly, therefore, in these engines the total weight per BHP averages about 325 lbs., the range being from about 275 to 385 lbs.

In the table on p. 742 a number of full-load trial results from four-cycle Diesel engines are exhibited in chronological order for purposes of reference.

It has been observed that as both the mechanical and brake thermal efficiencies of the Diesel are but little affected by the size of the engine, it is unnecessary, so far as considerations of fuel economy are concerned, to instal power in large units and distribute it by expensive transmission methods. Thus, the single-cylinder, 8 HP engine tested by Prof. Meyer had a mechanical efficiency of 79·4 per cent. and a brake thermal efficiency of 28·1 per cent., as compared with 78·3 per cent. and 30·4 per cent. respectively in the case of the single-cylinder, 80 HP engine tested by Mr. Ade Clark at Ghent.

A great deal of attention has in recent years been concentrated upon the Diesel engine, and the type is being gradually built in larger and larger power units. Up to the end of 1910, the largest output from a single-cylindered engine was about 1200 IHP; obtained by Messrs. Carels Frères from a single-acting, two-stroke Diesel having a bore of about 34¾ ins. and stroke about 47¼ ins., and running at 120 revolutions per minute. The piston was oil cooled, and the engine was of the cross-head type. The piston speed corresponding to 120

FOUR-CYCLE DIESEL ENGINES; FULL LOAD RESULTS

Date	Authority	Type of engine	Nominal BHP	Actual full BHP	Cyl. Diam. ins.	Piston stroke ins.	Revs. per min.	Piston speed, ft. per min.	Brake mean eff. pressure	Cooling water		Kind of fuel used	Lbs. fuel per BHP hour	Mechanical eff., per cent.	Brake thermal eff., per cent.
										Lbs. per BHP hour	Rise in temp. ° F.				
1897	Schröter	1 cyl. vertical	20	19·6	9·8	15·7	171·8	450	76·5	145·0	29·2	Kerosene	0·548	75·0	25·2
1898	Denton	1 cyl. vertical	70	65·2	16·0	24·0	160·0	640	66·9	—	—	Kerosene	0·534	69·0	25·7
1900	Meyer	1 cyl. vertical	30	29·7	11·8	18·1	181·2	545	65·8	—	—	Kerosene	0·457	76·5	30·0
1900	Meyer	1 cyl. vertical	30	29·9	11·8	18·1	181·2	545	65·8	—	—	Bavaria crude	0·477	74·0	29·8
1902	Meyer	1 cyl. vertical	70	68·6	15·75	23·6	158·8	625	74·5	18·6	118·0	Russ. kerosene	0·430	79·1	32·0
1902	Meyer	1 cyl. vertical	8	9·9	6·48	10·7	267·0	477	83·1	21·8	115·0	Russ. kerosene	0·489	79·4	28·1
1902	Ade Clark	1 cyl. vertical	35	39·2	11·8	18·1	182·5	550	85·9	45·0	55·3	Texan crude	0·461	75·0	28·9
1903	Ade Clark	1 cyl. vertical	80	79·5	15.75	23·6	160·0	630	85·6	—	—	Texan crude	0·434	78·3	30·4
1903	Ade Clark	2 cyl. vertical	160	164·8	15·75	23·6	154·5	607	92·3	19·4	94·5	Texan crude	0·406	80·7	32·6
1905	Longridge	3 cyl. vertical	500	459·0	22·05	29·52	152·8	752	70·7	22·0	78·7	Galician oil	0·451	77·0	31·7
1906	Am. Diesel Co.	3 cyl. vertical	225	232·0	16·0	24·0	162·0	648	78·3	—	—	'Distillate'	0·445	—	30·2
1907	Am. Diesel Co.	3 cyl. vertical	120	121·9	12·0	18·0	222·2	668	71·0	—	—	'Distillate'	0·470	—	28·5
1907	Barthe.	Otto-Deutz-Diesel	35	35·0	—	—	209·3	—	—	26·5	94·0	'Petroleum'	0·410	—	—
1908	Am. Diesel Co.	3 cyl. vertical	170	172·3	14·0	21·0	202·5	710	69·5	—	—	'Distillate'	0·460	—	29·2
1908	Ludwig (U.S.)	3 cyl. vertical	225	249·7	16·0	24·0	169·1	677	80·8	15·6	141·0	Desulphurised Texas fuel oil	0·464	—	28·1
1910	Mirrlees-Diesel	3 cyl. vertical	120	130·0	12·0	18·25	200·0	608	83·3	—	—	'Petroleum'	0·425	73·0	31·6

r.p.m. has the high value of 945 ft. per minute; at this speed, 1200 IHP requires an indicated mean effective pressure of 88½ lbs. per sq. in. Practical difficulties remain to be overcome with such large engines, which must accordingly be regarded as still in the experimental stage in some respects. A maximum compression pressure of from 450 to 500 lbs. per sq. in. is found sufficient to produce ignition even with the heaviest fuel oils, and is generally adopted. The mean effective pressure during the working stroke varies from about 80 lbs. per sq. in. in the largest engines to 115 lbs. per sq. in. in the smallest, the value being reduced in the larger sizes in order to keep temperatures from rising too high.

The preservation of gas-tightness in the piston has given much difficulty; the first Augsburg Diesel engine had a crosshead, and this is retained in the Riedinger design, and in several of the largest engines now being built; in the opinion of Guldner, Milton, and other authorities it is more satisfactory than the general practice hitherto adopted—largely for reasons of cost, size, and weight reduction—of causing the piston to take the side thrust of the connecting-rod; this practice necessitates the use of pistons of length about twice their diameter, and trouble has been experienced in ensuring adequate lubrication to prevent seizing. Some constructors coat the lower cylindrical surface of the piston with white metal to improve the running. The cross-head type appears likely to become the standard for large engines.

Cracked piston crowns through over-heating have also been numerous; it is this and seizing troubles that mainly limit the diameter of cylinders in practice. The fuel oil is injected vertically downwards upon the centre of the piston crown, and trouble has thus sometimes arisen; reference is made elsewhere to the employment of false crowns, or the insertion of a nickel plate in the crown, to minimise the effects of flame erosion. The maintenance of regular and rapid combustion has also proved a problem of difficulty; the central position of the injection valve has been adopted largely to promote a uniform distribution of the spray cloud throughout the combustion chamber space; many constructors so form the upper surface of the piston crown as to assist in the distribution of the spray cloud rapidly and evenly. Gudgeon bearing details have necessitated considerable attention, and a good form is that shown in fig. 447 (*infra*). The gudgeon is sometimes lubricated from the big end bearing by a duct carried along the shank of the connecting-rod; a better practice is to supply it with oil from the ring system of holes in the lower part of the cylinder barrel by which the piston is usually force-lubricated, through ducts drilled in the gudgeon pin and delivering the oil at the top of the bearing.

The consumption of lubricating oil in the modern Diesel need not exceed 2½ per cent. of that of the fuel oil.

Connecting-rods are in general of turned circular section in the steel shank, with built-up big ends of the standard marine type. Mr. J. T. Milton, M.Inst.C.E., has considered the actions occurring on the crankshafts of Diesel engines, and has been led to propose the following formula for their diameter:

$$\mathrm{D} = 0{\cdot}516 \sqrt[3]{d^2 s} \tag{6}$$

where D is the crankshaft diameter in the bearings, and d and s are the bore and stroke of the engine respectively, all in inches. This formula is appropriate to cases in which the distance from centre to centre of the crank bearings is about 1·2 times the length of stroke.

Crankshaft main bearings are very usually ring-lubricated and white metal lined, with a length ranging from one to two shaft diameters, depending upon the cylinder grouping. With shafts of diameter in accord with Eq. (6) and main bearings one diameter in length, the maximum pressure per sq. in. of projected bearing area does not exceed about 740 $\left(\frac{d}{s}\right)^{\frac{2}{3}}$ lbs. per sq. in. in normal full-load running; for present Diesel engines $\left(\frac{d}{s}\right)$ varies but little from $\frac{2}{3}$; hence the maximum bearing pressure is about 560 lbs. per sq. in.; in marine steam-engine practice pressures from 400 to 600 lbs. per sq. in. are employed; the figures are thus concordant.

Piston speeds range usually from 650 to 750 ft. per minute. The ratio of connecting-rod to stroke varies from about 2¼ to 2¾.

The mechanical efficiency of the four-cycle type in recent examples is from 75 per cent. to 80 per cent.; the brake thermal efficiency with this type averages about 30 per cent., corresponding to an average indicated thermal efficiency of about 38½ per cent. Brake thermal efficiencies rather over 32 per cent. and indicated efficiencies rather over 40 per cent. have been recorded.

Engines tested in 1900 and 1902 by Prof. Meyer had volume ratio of compression 16·3 and 15·4 respectively; a Carels Diesel engine tested by Mr. Ade Clark in 1902 had a ratio 14·3; in present practice the ratio appears to be between 14 and 15.

The air standard efficiency corresponding to a volume ratio of compression of 14 is 0·659; a 40 per cent. indicated thermal efficiency corresponds therefore to a percentage relative efficiency of about 61 per cent.; the four-cycle Diesel engine is the most economical in fuel consumption that has so far been produced.

Two-stroke Diesel Engines.—Messrs. Sulzer Bros., of Winterthur, have for some years taken a leading position in the development of

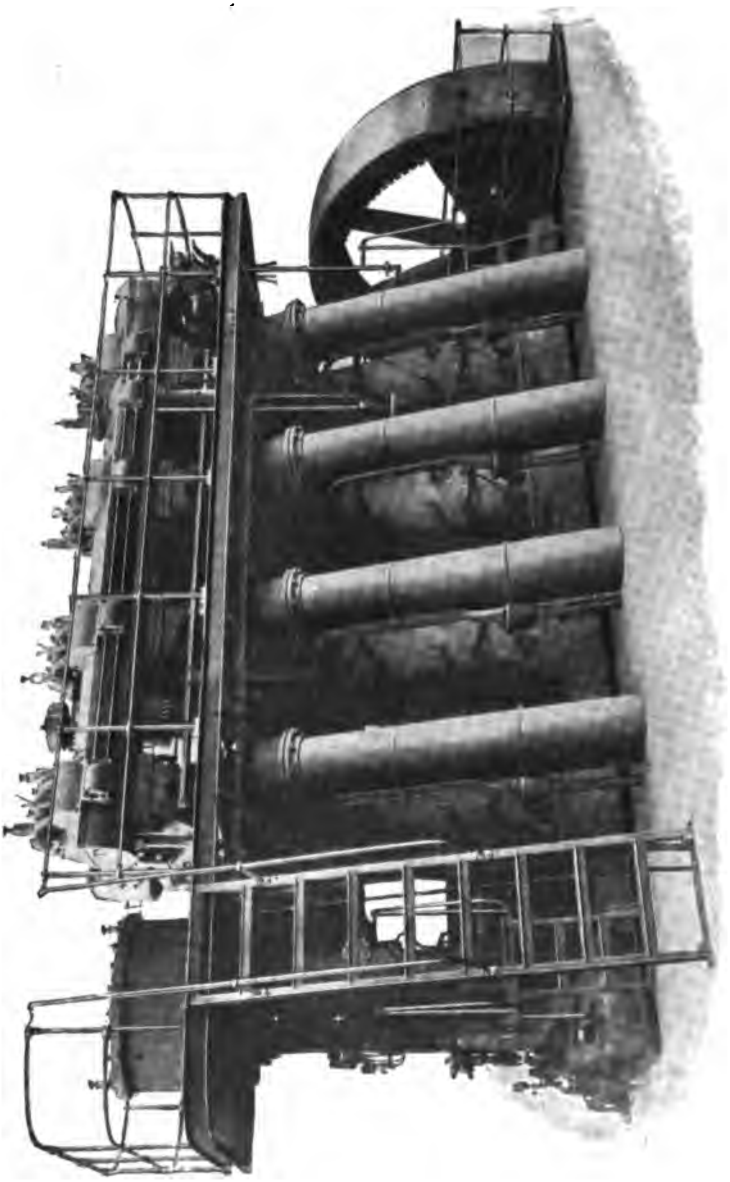


FIG. 445

this type of engine; in the marine section of the Milan Exhibition of 1906 a two-stroke, four-cylinder, 100 BHP reversible Sulzer-Diesel engine was shown.

An illustration of the four-cylinder, two-stroke, 1000 BHP land type, as shown at the Turin Exhibition in 1911, is given in fig. 445; the normal speed is 150 revolutions per minute. In fig. 446 a transverse section is shown through one of the cylinders; the oil-cooled piston, with six spring rings in the upper part and two oil-excluding rings in the lower, will be noted. As usual, the fuel injection is in the centre of the cylinder head, together with the air and starting valves.

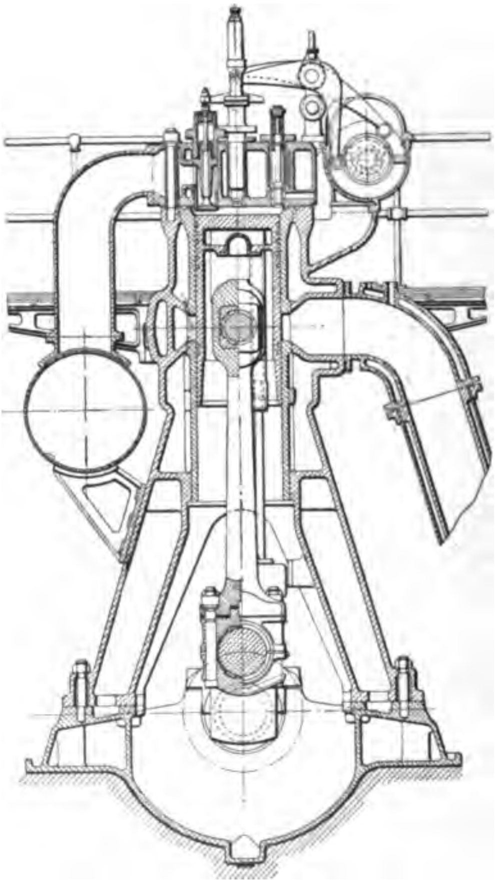


FIG. 446

During 1911 a four-cylinder, two-stroke, single-acting engine of this type of 2400 BHP was installed in an electric generating station in France.

Messrs. Krupp have also constructed a large number of single-acting, two-stroke Diesel engines at Kiel, while the Augsburg-Nürnberg Company build large double-acting, two-stroke Diesels of both vertical and horizontal type. In fig. 447 transverse sections are shown of the four-cylinder, two-stroke, single-acting reversible marine Diesel engine giving 25 BHP per cylinder at 375 r.p.m.; the bore is 7·1 ins., and the stroke 9·85 ins. The piston speed corresponding to 375 r.p.m. is 615 ft. per minute, and brake MEP about 68 lbs. per sq. in. The action is as follows: Near the end of the working stroke the piston overruns the large exhaust ports E E located in the lower part of the cylinder barrel, through which the burnt gases are discharged; immediately after this, air is admitted to the cylinder through the cam-operated air valves A A in the head at a pressure of about 5 lbs. per sq. in.; this scavenges the cylinder and hastens the expulsion of most of the remaining exhaust. The return of the piston closes the ports, and the air valves in the head being then at once closed the entrapped air in the cylinder is compressed into the combustion chamber, when

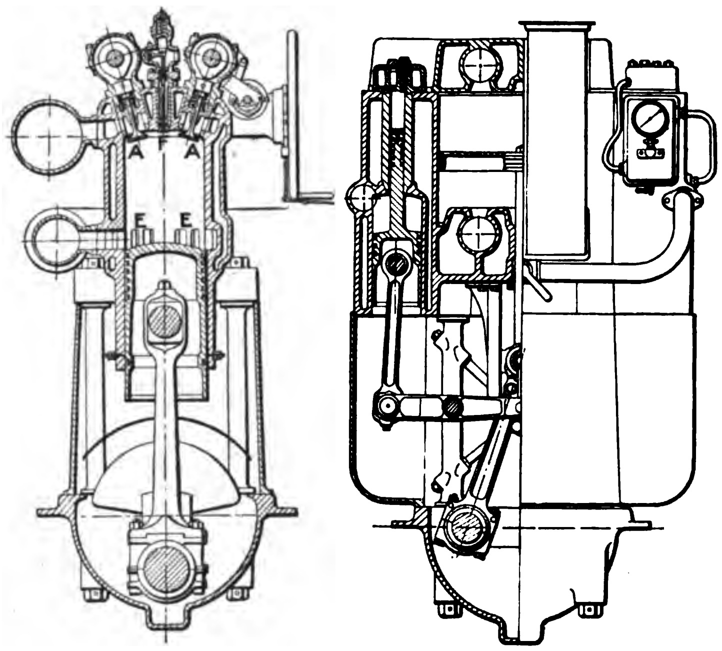


Fig. 447

injection of oil and inflammation of the mixture follow as usual; thus every downward stroke of the piston is a working stroke. Messrs. Sulzer are experimenting with designs in which the cam-operated air valves in the cylinder head are omitted, the air entering the cylinder through ports on one side which are overrun by the piston, while the exhaust ports are located on the other side of the cylinder. Reverting to fig. 447, F is the oil fuel needle valve; the compressed air starting valve is also located in the cylinder head, but does not show in this sectional view. The piston has six spring rings and a seventh oil-retaining ring near the bottom; piston lubrication is forced through the usual ring of holes in the lower part of the working barrel. The gudgeon end is solid with the connecting-rod, and is fitted with adjustable brasses held together by a stout set screw on top. The cylinder is supported from the bed-plate on turned steel stanchions, while covering doors are fitted so that the crank-chamber is enclosed. In the right-hand view are shown the two-stage water-cooled air compressing pump, and also the large scavenging air pump in half section with its water-cooled cylindrical Gutermuth valves at top and bottom; the high pressure air compressing pump takes its air from the delivery side of the scavenging pump and discharges into the compressed air reservoirs, whence the blast is taken for the oil injection; in marine service, where there is much manœuvring, these air reservoirs can be kept charged by an auxiliary separately driven compressing pump to prevent any risk of failure of the supply. In the two-stroke Diesel engine, therefore, a large scavenging air pump is necessary to supply air at low pressure to the cylinders in addition to the high pressure compressor required for the fuel oil blast and for starting, reversing, and general manœuvring purposes.

For marine service the two-stroke engine possesses the important advantages of: (1) greater uniformity of torque per cylinder; (2) being more readily arranged to be reversible than the four-stroke engine. With four-cycle engines ability to reverse involves in general somewhat complex and cumbrous additions to the valve gear, and hence it is common with such engines to interpose some form of mechanical or electrical reversing device between the engine and the propeller shaft.

Owing to the absence of exhaust valves the two-cycle engine can be more simply arranged to reverse, and may be coupled directly to the propeller shaft; it must be remembered, however, that with these engines also the phase of the camshaft actuating the several valves must be altered when reversal is desired.

To start the engine the blast air and oil valves are cut out of action, and the compressed air starting valves alone operated by suitably moving the hand-regulated wheel which controls the engine. The two-cycle marine engine has very usually four cylinders, and the cranks are placed at right angles so as to obtain a regular sequence of working impulses ; sometimes six, and occasionally eight, cylinders are found. Hence, taking for example a four-cylindered engine, one compressed air starting valve must be open, and the engine accordingly moves off ; after one or two revolutions have occurred a further movement of the hand wheel cuts off the compressed air from two of the cylinders and brings the combustion valves of these into action ; these two cylinders at once take up their normal working cycle, the remaining two meanwhile maintaining the motion under the action of the compressed air. A third and final movement of the hand-wheel cuts off the compressed air and brings into play the combustion valves of these last two cylinders, which, in their turn, now take up their working cycle, all four then becoming power producing.

One mode of actuating the valves is indicated diagrammatically in fig. 448 ; projections A and B formed on the strap of an eccentric sheave C carried on the camshaft D form the ahead and astern driving cams respectively, actuating the valve through the contact roller R. When the fulcrum F is in one position the ahead cam alone is operative ; when moved into some other position, as F′, the astern cam actuates the valve ; for a position intermediate beween F and F′ neither cam operates, and the valve is then out of action.

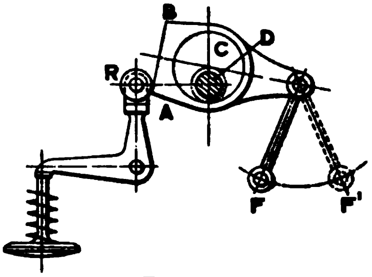


FIG. 448

To reverse from ahead running the hand wheel is first rotated back to the position in which *all* valves are non-operative ; a further rotation of the hand wheel in the reverse direction shifts the fulcrum F towards the position F′, thus bringing the air starting valves into proper action for reverse running. Motion of the engine in the reverse direction at once commences. Further movement of the hand-wheel in the reverse direction brings the firing cycles of the cylinders into action just as when starting for ahead running. The whole operation of reversal from full speed ahead is effected easily within five seconds, the movement of the hand wheel from ahead to astern being almost continuous.

In common with all two-stroke engines the scavenging is not quite complete, and hence the full benefit of the impulse every revolution is not realised ; some loss of compression may occur also in slow running engines by leakage past the topmost piston rings for a short period after the closing of the exhaust ports by the piston. Moreover the mechanical efficiency of the engine is necessarily low owing to the use of a large scavenging air pump.

The oil consumption per BHP hour is accordingly larger than in the four-cycle type by probably about 10 per cent in general.

The oil is injected vertically downwards towards the centre of the piston crown, and, due to the number of firing strokes being twice as numerous as in the four-cycle type, there is more likelihood of trouble from cracked crowns and seized pistons in two-cycle engines. Messrs. Sulzer let a nickel plate into the piston crown where the fuel flame impinges; some constructors employ false crowns, which can be readily replaced if they become burnt out.

Notwithstanding these drawbacks, the more uniform torque resulting in reduction or even elimination of the flywheel, and the comparative ease of reversal, have caused the two-cycle Diesel engine to be generally preferred in marine applications. Extracts from a paper by Mr. J. T. Milton, M.Inst.C.E., on the Diesel engine considered with reference to marine service will be found in the succeeding chapter of this volume. He concludes that the Diesel marine engine should be 'Diesel' only so far as the cylinders and their accessories are concerned, and standard marine engine practice in all other respects.

Owing to the comparatively recent introduction of the two-stroke Diesel as a commercial engine, but few systematic trials have been made; the authors are indebted to the Diesel Engine Company, of London, for the following results of test of a single-cylinder, two-stroke Diesel engine. The bore was 12·2 ins. and stroke 18·1 ins.; the mean speed was 205 revolutions per minute corresponding to a piston speed of 620 ft. per minute. Texan fuel oil was employed; this has a sp. gr. of 0·925, and a calorific value about 19,200 B.Th.U. per lb. The test results are tabulated hereunder, and fig. 449 is a chart exhibiting the mode of variation of several of the quantities.

TEST OF TWO-STROKE 12·2 IN. × 18·1 IN. DIESEL ENGINE

Revs. per min.	Horse-Power		Per cent. mech. eff.	MEP, lbs. per sq. in.		Lbs. oil per hour		Thermal eff.	
	Indic.	Brake		Indic.	Brake	per IHP	per BHP	Indic.	Brake
206·3	75	40	53·3	68·0	36·2	0·320	0·6	41·5	22·1
207·0	85	50	58·8	76·8	45·2	—	—	—	—
206·3	95	60	63·2	86·0	54·3	0·354	0·56	37·5	23·7
204·6	105	70	66·7	96·2	64·0	—	—	—	—
202·5	115	80	69·6	106·0	73·8	—	—	—	—
201·0	120	85	70·8	111·5	79·0	0·354	0·5	37·5	26·5

An illustration of a recent design of four-cylinder, two-stroke, single-acting vertical engine of the crosshead type, by the Diesel Engine Co., Ltd., of London, is given in fig. 450. This is of 1000 BHP, with cylinders about 19¾ ins. bore and a stroke of 26½ ins.; the piston is oil cooled, and the general arrangement of the crosshead is clearly indicated. The air scavenging pump is shown in section at the right-hand end of the longitudinal view.

Semi-Diesel Engines.—The high compression pressure necessary to ensure regular and rapid ignition in the Diesel engine necessarily calls for extreme care and skill in design and construction, which results in somewhat high production cost; moreover, as designers are practically limited to cast iron as the material for cylinders and certain other parts, the engines are rather heavy. It is also probable that the physical properties of cast-iron impose a limit of a quite moderate diameter upon the practical dimensions of cylinders; to safely withstand even 500 lbs. per sq. inch, independently of any necessity for providing against occasional pressures considerably in excess of this, soon involves cylinders of such great thickness that the preservation of a sufficiently low temperature of the inner walls may become difficult, apart from any considerations of strength.

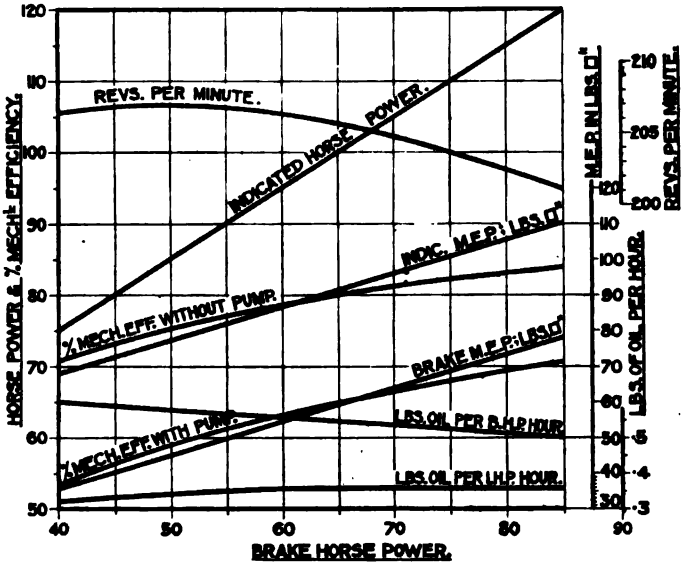


Fig. 449

Accordingly many attempts have recently been made to retain the high fuel economy of the Diesel without recourse to so high an initial pressure, and without much sacrifice of mean effective pressure;

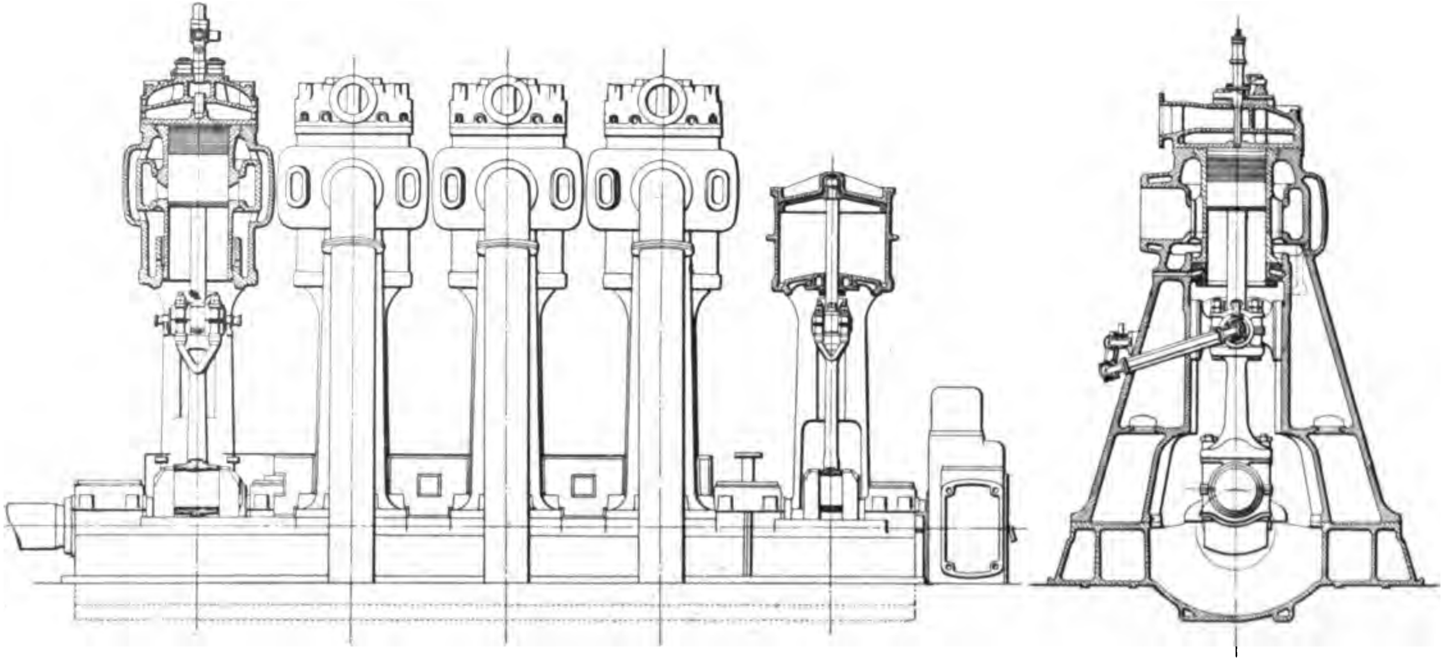


FIG. 450

in some cases also attempts have been made to dispense altogether with the very high pressure air blast, which has proved a source of serious trouble on many occasions. Considerable success has already been attained in this direction and, as will be seen directly, certain engines of the 'semi-Diesel' type have shown on test a fuel consumption as low as 0·45 lb. per BHP hour only. This is a very marked advance upon early oil engine practice; for example, in the 1894 R.A.S.E. trials at Cambridge (*v.* table p. 690) the best result was 0·82 lb. and the highest 1·68 lbs. per BHP hour.

An engine that has aroused some interest, and which may be described as a Hornsby-Diesel, is that manufactured by the De la Vergne Machine Company, of New York; this is a four-cycle engine wherein air only is compressed to about 200 lbs. per sq. in.; for ignition a hot bulb is relied upon into which the fuel oil is sprayed near the end of the compression stroke by an air blast at a pressure of some 600 lbs. per sq. in. only; the air blast is provided by a very small two-stage direct-acting compressor attached to the side of the engine cylinder and eccentric-driven from the main shaft.

The fuel oil is delivered to the air spray nozzle by a pump, and governing is effected by varying the length of stroke of the pump plunger; the compression is thus constant, and the fuel consumption per horse-power hour varies but little from full to half load, or even less. As in the Diesel, the air blast compressor stores air in a reservoir, and this is employed for starting purposes.

The makers state that oil consumption as low as 0·4 lb. per BHP hour has been several times recorded. These engines are built in sizes from 85 to 175 BHP with single cylinders, and in two-cylinder form from 170 to 350 BHP; it will be noted that they are Diesel in all respects excepting that the mode of ignition is that of the Hornsby-Akroyd type; the De la Vergne Company has for some years manufactured these latter engines in the United States, and the type just described is accordingly a product of their experience with the two systems.

Among English firms who are now also building so-called 'semi-Diesels' may be mentioned the Blackstone Co., whose crude oil engine is described in Chap. XI, the National Gas Engine Co., Petters, Ltd., and Ruston, Proctor & Co., Ltd. Messrs. Ruston's design is of special interest inasmuch as the air blast is dispensed with altogether. The engine operates on the four-stroke cycle, air alone being compressed into the combustion chamber and hot bulb by which ignition is effected; a section through the breech end of the cylinder is given in fig. 451. Nearly at the instant of maximum compression the oil fuel is injected by a force feed oil pump operated by a quick-acting cam through the mechanical atomiser indicated, which consists essentially of a number of small ducts converging to a central inlet orifice from which the oil accordingly issues in the form of a fine cloud of spray into the hot bulb; the oil inlet valve is spring-loaded and is lifted by the pumped charge of oil at each working stroke; the lift of this valve is only about 0·025 of an inch.

A general external view of the 80 BHP Ruston 'C.C.' oil engine is given in fig. 452. The C.C. engines are built in single-cylinder sizes from 20 BHP at 240 r.p.m. to 80 BHP at about 190 r.p.m.; and of double-cylinder type from 80 to 130 BHP at about 200 r.p.m. Water injection is employed for three-quarter load and above to ensure smooth running.

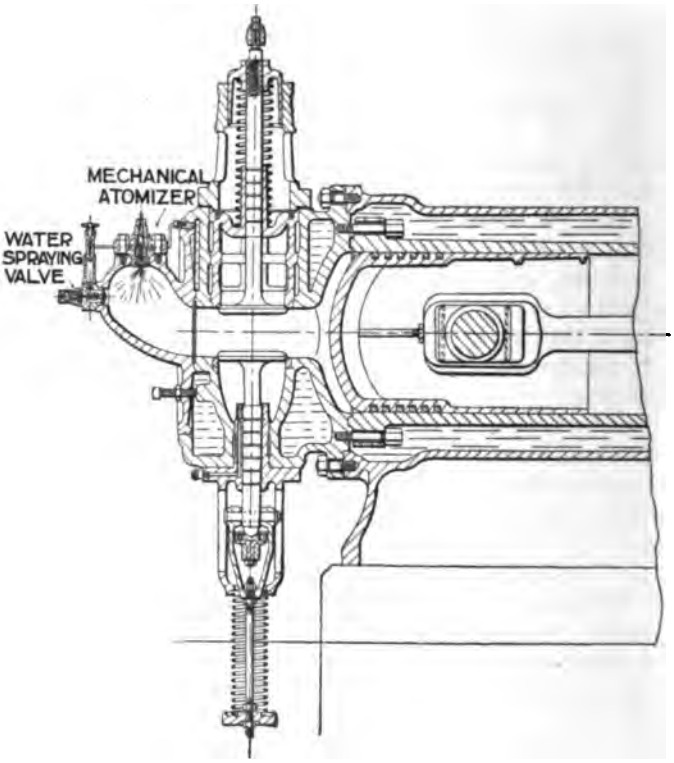


FIG. 451

The crankshaft and camshaft are ring lubricated; the big end is centrifugally lubricated; forced lubrication is provided for the piston, gudgeon, and exhaust valve spindle. These engines are governed by by-passing some of the fuel oil delivered by the pump, and returning it to the oil reservoir.

The compression pressure employed is about 275 lbs. per sq. in., and the maximum explosion pressure about 400 lbs. per sq. in. only; at full load, mean effective pressures as high as 95 lbs. per sq. in. are obtained; the ratio of this MEP to the maximum pressure is thus about 1 : 4·7.

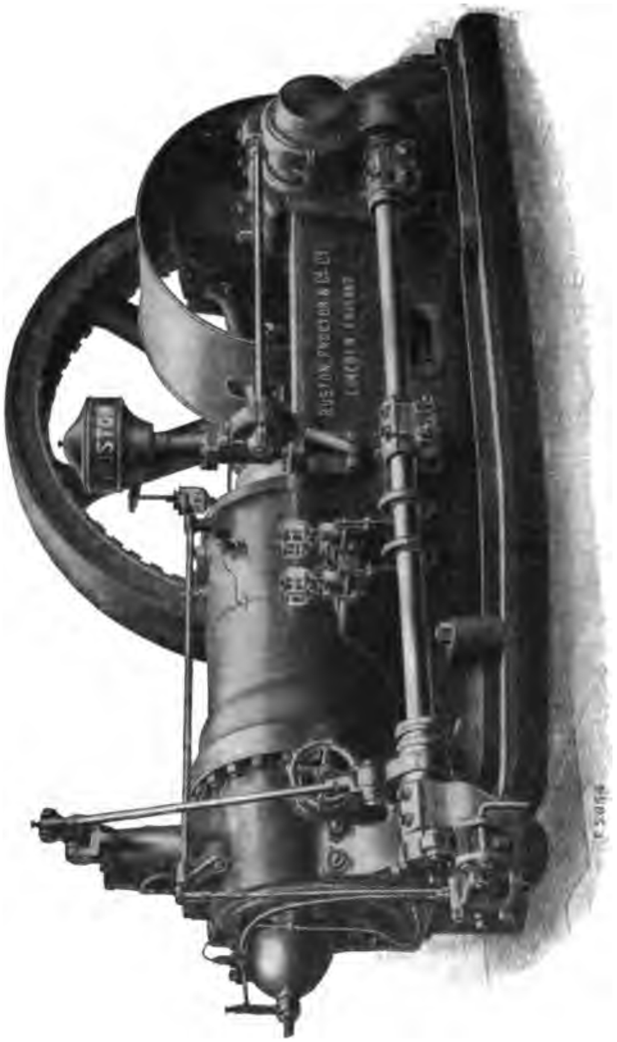


Fig. 452

The weight of the single-cylinder, 50 BHP 'industrial' design is about 7¼ tons, corresponding to 320 lbs. per BHP. On December 1st and 2nd, 1910, tests were carried out at Lincoln by Prof. W. Robinson on a 50 BHP horizontal engine of this type; the cylinder bore was 14 ins. in diameter, and the stroke 20 ins.; the normal speed was 205 r.p.m., corresponding to a piston speed of about 685 ft. per minute. The engine was started by compressed air; the time occupied in starting from cold, due to the necessary lamp-heating of the hot-bulb vaporiser, was 12 minutes; the engine was then running with load. The test was commenced five minutes later, i.e. 17 minutes from 'all cold.' The load was imposed by a friction brake applied to the engine flywheel.

To test the governing, after the full load trial on December 1st, the load was suddenly removed, when the speed of the engine increased from 206 to 212 r.p.m. for a few moments, settling down steadily to a speed of 208 r.p.m.

TRIALS OF RUSTON CRUDE OIL ENGINE BY PROF. W. ROBINSON.
DEC. 1910

Date of trial	Dec. 1, 1910	Dec. 2, 1910	Dec. 2, 1910
Load on engine	full	full	three-quarter
Kind of oil used	Russian crude	Italian refuse	Italian refuse
Duration of trial, hours	2·0	2·0	1·0
Mean revolutions per minute	205·7	205·5	208·8
Indicated horse-power	66·3	64·1	51·8
Brake horse-power	51·8	50·8	38·5
Mechanical efficiency, $\frac{\text{BHP}}{\text{IHP}}$, per cent.	78·0	79·3	74·5
Total oil used per hour, lbs.	23·25	24·90	18·0
Oil per IHP hour, lbs.	0·35	0·388	0·347
Oil per BHP hour, lbs.	0·45	0·49	0·468
Thermal Data:			
Cal. value (lower) of fuel, B.Th.U. lbs.	18,000	17,600	17,600
Indicated thermal efficiency, per cent.	40·4	37·4	41·7
Brake thermal efficiency, per cent.	31·4	29·5	30·8

The vaporiser was maintained by the successive explosions at a dark heat, never approaching dull red; the ignition on compression was regular and automatic. Prof. Robinson states in his report on this test that the indicator diagrams showed 'perfectly regular ignition of the oil spray in the compressed air at 270–280 lbs. per sq. in., and that the highest pressure of explosion was from 360 to 395 lbs per sq. in. At full load the MEP varied from about 85 to 82 lbs. per sq. in. with Russian crude petroleum, and was about 80 lbs. per sq. in. with Italian refuse oil.' In the trial of December 1st Russian crude oil was used; this had at 60° F. a specific gravity of 0·875 and a calorific value in B.Th.U. per lb., as determined by a Mahler-Cook bomb calorimeter, of 19,100 higher value, and 18,000 lower value. On December 2nd an Italian refuse oil was used of sp. gr. 0·947, higher heat value 18,620, and lower 17,600 B.Th.U. per lb.

The results obtained are tabulated on p. 756.

These results compare favourably even with the best Diesel practice; Prof. Robinson states that the engine ran throughout the trials with extreme steadiness.

CHAPTER XI

MARINE GAS AND OIL ENGINES

THE extent and importance of inland and coast water transport, especially before the great modern development of railways, necessarily soon attracted the attention of inventors of internal combustion engines to the problem of boat propulsion by this means, and early attempts are referred to in the historical sketch in Vol. I of this work. There will be found reference to the boat of Samuel Brown which ran on the Thames in 1827; to Lenoir's small craft which later plied between Paris and Charenton; to Brayton's boats fitted with horizontal engines which ran on the Hudson river; and to Daimler's small fast-running petrol engines fitted to small launches which have been so largely used in Germany, Holland, Belgium, and France. At the Paris Exhibition of 1889 one of these small boats appeared on the temporary lake near the Pont d'Iena, and it was on this that the first meeting took place of MM. Daimler and Levassor, whose association contributed so greatly to the development of modern automobilism.

Yarrow's early spirit launch 'Zephyr' and the 'Alco-vapour' boats of the Marine Engine Co. of Harrison, N.J., were not of the internal combustion type; in these, petrol and alcohol respectively were used to replace the water ordinarily employed in the boiler; this type has not survived.

In the United States, with its many magnificent lakes, rivers, and sheltered waters, the petrol or 'gasolene' motor-boat has for long been very popular and is largely used both for pleasure and commercial purposes; petrol is almost universally employed as the fuel, partly on account of the long familiarity of the people with it in domestic use for cooking, &c., and partly because of its cleanliness and cheapness in comparison with kerosene. Though the two-cycle motor of the type invented by Day in 1891 has not hitherto found great favour in England, it largely predominates in the States; this is due in part to its cheapness and the simplicity arising from the absence of valves, and in part to the early adoption of electric ignition by the Americans.

When first introduced in England ignition by hot tube was in general use, and accurate timing of the explosions was hence difficult, which resulted in these engines manifesting a tendency to occasionally reverse suddenly; in the United States electric ignition was adopted earlier, and the instant of firing could with this be quite accurately determined.

The prevailing American type is the simple 'two-port' engine in which the charge of carburetted air is drawn through a non-return inlet valve into the crank-chamber during the up-stroke of the piston, and compressed therein to 3 or 4 lbs. per sq. in. above atmosphere on the down-stroke; when near the bottom of its stroke the upper edge of the piston first overruns an exhaust port and almost immediately afterwards uncovers an inlet port in communication with the crank-chamber; the entering charge, being under slight pressure, assists in the expulsion of the burnt gases. On the next up-stroke the charge is compressed into the combustion chamber, fired, and the usual working stroke follows. It will be noted that there are neither inlet nor exhaust valves, and also that the engine possesses the advantage that it will run indifferently in either direction. In general the fuel consumption of these engines is somewhat high,

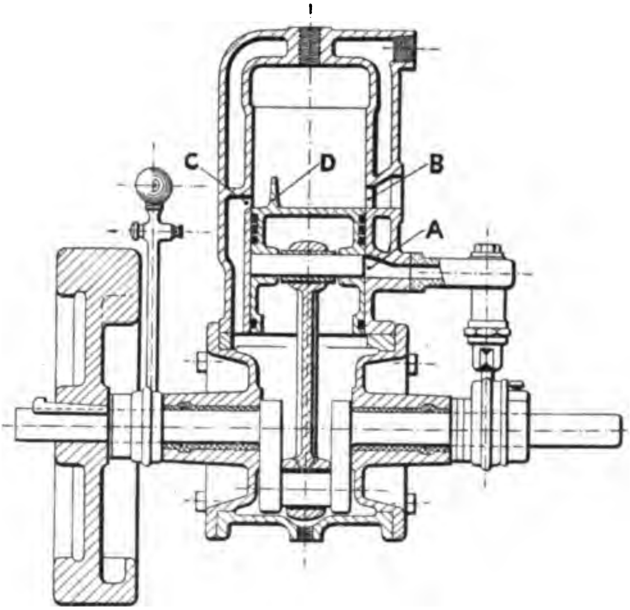


FIG. 453

due to the escape of a portion of each fresh charge through the exhaust port.

A type also used largely in American motor launches is the 'three-port' engine illustrated in fig. 453; this is of a 3¼ in. × 3¼ in. marine Day engine, rated as of 2½ horse-power at 900 revolutions per minute. The ascent of the piston causes a partial vacuum in the crank-chamber, and when near the top of its stroke its lower edge uncovers the port A. thus permitting carburetted air to rush into the crank-chamber. The descent of the piston closes this port and then compresses the charge in the crank-chamber to 3 or 4 lbs. per sq. in. above atmosphere; when near the bottom of its stroke its upper edge first uncovers the exhaust port B and next the inlet port C; the lip D on the piston deflects the entering stream upwards so as to prevent as far as possible any 'short-circuiting' of the fresh charge through the exhaust port. In the first three-port engines no valves of any kind were used, but later it was found to be advantageous to provide a non-return inlet valve in the suction pipe, as indicated in fig. 453.

Leakage from the crank-chamber is commonly prevented by making the crankshaft bearings very long and using grease as a lubricant. Experiments with an early model of the engine shown in fig. 453 were made by Professor W. Watson, F.R.S., and Mr. R. W. Fenning, B.Sc., in 1910. It was found that the proportion of each fresh charge which escaped unburnt through the exhaust port was considerable at low speeds, but diminished with increase of speed as indicated hereunder:

Revolutions per minute	Per cent. of fresh charge lost through exhaust port
600	36
1200	20
1500	6

The mean effective pressure was much higher at low than at high speeds, ranging from about 62½ lbs. per sq. in. at 600 r.p.m. to 44¼ lbs. per sq. in. only at 1500 r.p.m.; the volumetric efficiency was about 40 per cent., and varied but little with speed, the greater loss of the fresh charge through the exhaust port at low speeds approximately counterbalancing the larger volume of charge then entering the cylinder.

At high speeds considerable wire-drawing occurred between the carburettor and crank-chamber; to lengthen the duration of the communication between them, ⅛ in. was cut off the lower edge of the piston, opposite the port; much improved results were thus obtained, particularly at high speeds, the mean effective pressure being well maintained, as the following figures show:

Revs. per min.	Mean effective pressure, lbs. per sq. in.	Horse-power		Per cent. increase in IHP due to increase in port
		Indicated	Brake	
600	57½	2·35	1·71	− 8
900	65½	4·02	3·27	12
1200	63½	5·2	4·2	21
1500	54½	5·57	4·4	19

While the IHP was by this means materially increased at all speeds above the lowest, the gross efficiency was reduced by the increased loss of charge through the exhaust port.

Owing to the large admixture of exhaust products with the fresh charge the mixture for this two-cycle engine required adjusting within comparatively narrow limits in order to obtain regular ignition; otherwise ignition only occurred at every alternate out-stroke, the intermediate stroke having a scavenging effect only; with this alternate firing, due to the richer mixture, much higher explosion pressures occurred, causing correspondingly increased engine stresses which, the experimenters suggest might be dealt with in a lightly designed engine by providing a relief valve in the cylinder head. The ideal two-stroke engine would yield twice as much power at the same speed as a four-cycle engine of equal bore and stroke, but due to imperfect scavenging, loss of fresh charge through exhaust, and wire-drawing during admission with consequent low volumetric efficiency and increased dilution of charge, the two-stroke petrol engine of this general type usually shows in practice only a small advantage in point of power.

In this instance Professor Watson and Mr. Fenning concluded from a comparison with the performance of a Siddeley and a Clément-Talbot engine tested some time previously that the Day engine gave about 47 per cent. excess power at 900 r.p.m., and about 29 per cent. excess at 1500 r.p.m. of that of the corresponding four-cycle engine. The comparison was made by doubling the value of the mean effective pressure given by the two-stroke engine, and taking the ratio of this to the mean effective pressure as shown by the four-cycle engines. These two four-cycle engines were not, however, of recent design; present-day well-designed petrol engines of 3¼ in. bore—though with longer stroke than that of this two-cycle engine—will give a value of ηp of 90 lbs. per sq. in. without difficulty. Now even after removing ⅛ in. from the lower edge of the piston, the highest value of ηp attained in the Day engine (at 900 r.p.m.) was 53¼ lbs. per sq. in., and doubling this we obtain 106·5, the ratio of which to 90 is 1·18 only; thus, estimated on this basis, the performance of this early two-stroke engine was, at best, only 18 per cent. greater than that of a corresponding recent four-stroke engine.

In the small four-cycle petrol engines of cars increase of power has been attained largely by increase of speed; but for marine (and aeronautical) purposes it is important to be able to obtain increased power without increasing revolution speeds, and accordingly it is in such cases that an efficient two-stroke engine will prove of great practical value; in single-cylinder boat motors also less vibration occurs with the two- than with the four-cycle type.

Many imitations and modifications of the original Day engine have been made, especially in America, as for example in the well-known marine motors of the Lozier Co., of Plattsburg. Mention may also be conveniently made here of the recent very successful application of this type to the motor bicycle in the case of the little two-cylinder 2⅞ in. × 2½ in., 3¾ HP, *water-cooled* engines of the Scott Engineering Co., Ltd., of Bradford.

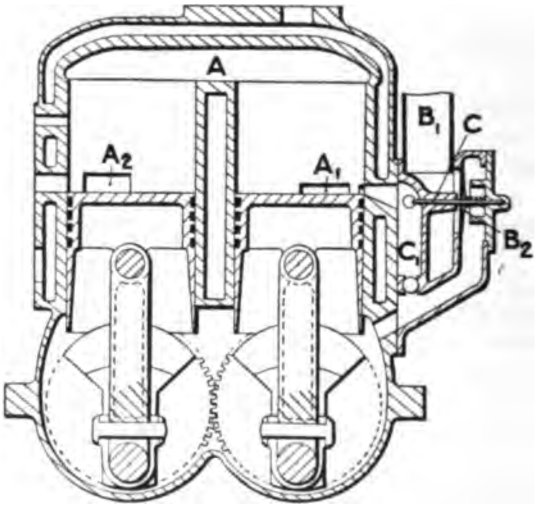


FIG. 454

In the well-known 'Valveless' engine of Mr. R. Lucas one long cylinder was originally employed, with two pistons moving in opposite directions, but later he adopted the so-called 'syphon' arrangement illustrated in fig. 454. The cylinders are cast in pairs with a common combustion chamber, connection being made by the large port A between them; in one cylinder the inlet port A_1 is formed, and in the other the exhaust port A_2, these being respectively overrun by the two pistons, which move together; there are two crankshafts geared together as indicated. The loss of fresh charge through the exhaust port which is a prominent defect of the ordinary single-cylindered two-cycle engine, was hoped to be avoided in this design by the baffling action of the high cylinder wall interposed between the ports A_1 and A_2; but in this, as in the ordinary type, the exhaust port does not close until after the inlet has closed, and some loss of charge is still accordingly experienced, notwithstanding the distance separating the ports.

Carburation is effected by a mixing valve shown in section in fig. 454; the crank-case suction induces a supply of air through the main inlet B_1 by way of the large spring-controlled disc valve B_2; in rising from its seat B_2 engages with a collar on the spindle of the small needle valve C, which is thus also lifted, permitting petrol to flow from a small orifice into the mixing chamber C_1; it will be noted that only the air from B_1 enters the crank-chamber, the petrol spray and vapour remaining mainly in C_1 until the inlet port is opened by the piston in its descent. Obviously this engine will run equally well in either direction; in order to reverse, the speed is reduced as much as possible and the ignition then suddenly advanced; firing occurs too early, and the pistons are checked, stopped momentarily, and their direction of motion immediately changed; the ignition is then set at its normal position for running. The 20 HP engine comprises two cylinders as shown, each $5\frac{1}{4}$ ins. in bore and with a stroke of $5\frac{1}{2}$ ins.; a power-speed graph is shown in fig. 455; the full horse-power is attained at a piston speed of about 900 ft. per minute, with a maximum of roundly 21 horse-power at 1100

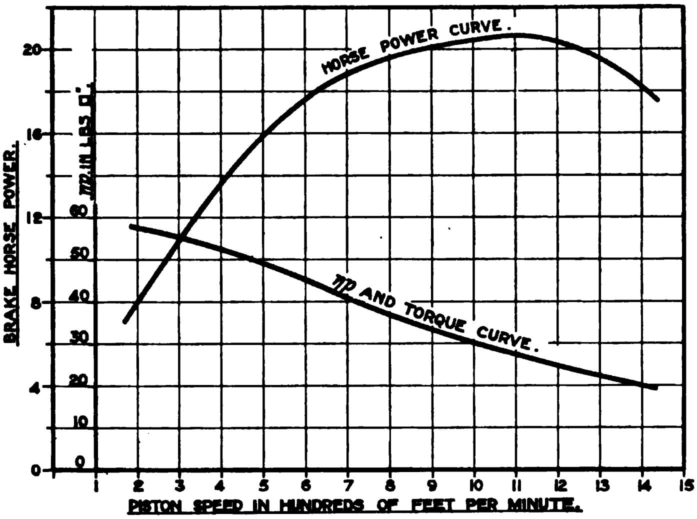


FIG. 455

ft. per minute. The curve of brake mean effective pressure, ηp, showing also the torque variation, shows that the latter continuously increases with reduction of engine speed. At the low speed of 180 ft. per minute ηp has the value 58½ lbs. per sq. in., while at 900 ft. per minute, when full power is developed, ηp is only 33½; if we double this for comparison with a four-cycle engine of equal cylinder diameter, we obtain the figure 67, which is low; there is no great difficulty in obtaining a value of upwards of 90 lbs. per sq. in. for ηp at full load from a 5½ in. four-cycle cylinder. The values of ηp from this test are, in fact, of the same order as those obtained by Prof. Watson and Mr. Fenning from the three-port Day engine previously described. In the two-stroke vertical Lamplough engine both pistons are connected to one and the same crank-pin as indicated in fig. 456, and thus arranged the two pistons are not quite in the same phase throughout their motion, with the result that, for equal width of opening, the exhaust port opens first and closes first, thus overcoming the objection that in the usual two-stroke arrangement the exhaust closes after the inlet.

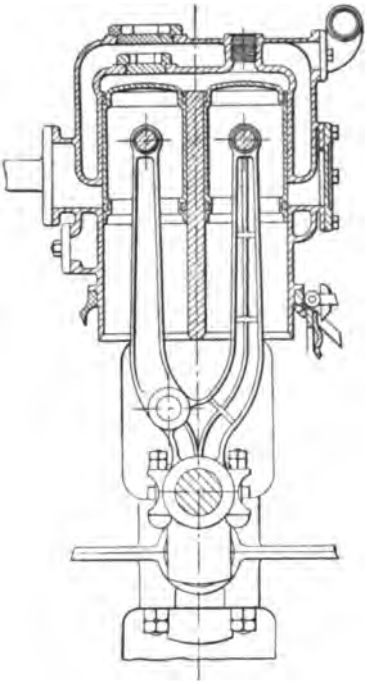


Fig. 456

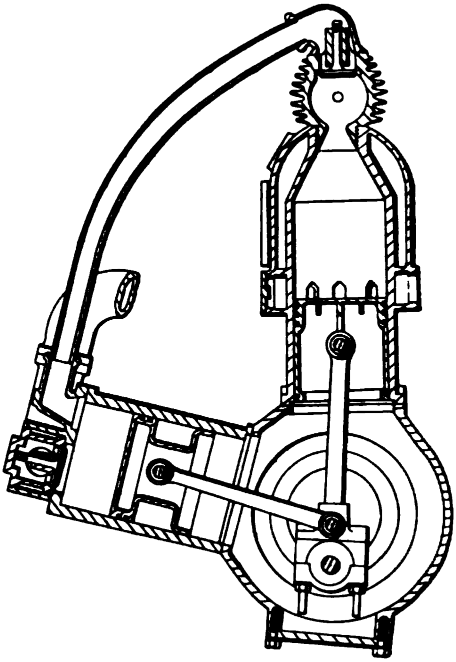


Fig. 457

Mention may here conveniently be made of the recently introduced Dolphin petrol engine, which works on the two-cycle method invented by Clerk in 1881 (*v.* Vol. I, p. 34); a sectional illustration is given in fig. 457. In the Dolphin engine the cylinder head is furnished with an air-cooled bulb below the automatic inlet valve into which the fresh charge from the displacer first enters; the functions of this bulb are: (1) To ensure that at all times there shall be an undiluted and readily ignitible mixture in the vicinity of the sparking-plug; and (2) To diminish the turbulence of the entering mixture and tend to cause it to advance and expand more uniformly along the conical combustion head shown, and thus cause the more effectual expulsion of the burnt gases without itself suffering loss through the exhaust port. Tests with a two-cylinder engine of this type are said to show that at no load steady running and regular firing are obtained at the low rate of 130 revolutions per minute. In the following tables some test results are given from a two-cylinder, 3¾ ins. × 5 ins. Dolphin engine; the displacer pistons were each 4¾ ins. bore with 4 ins. stroke.

TEST OF TWO-CYLINDER DOLPHIN ENGINE AT CONSTANT SPEED AND VARYING POWER

Revs. per minute	BHP	Pints of petrol per BHP hour
1000	5·0	1·28
1000	9·5	1·05
1000	11·6	0·91
1000	14·0	0·82
1000	18·0	0·78
1000	19·1	0·75
1200	21·5	0·71

Taking the calorific value of the petrol as 16,800 B.Th.U. per pint, 0·71 pint corresponds to a brake thermal efficiency of 21·3 per cent. In the following table the power was varied by varying the speed:

Revs. per minute	Piston speed, σ, ft. per min.	Horse-power		Mechanical efficiency, η, $\frac{\text{BHP}}{\text{IHP}}$	Indicated MEP, p, lbs. per sq. in.	Brake MEP, ηp, lbs. per sq. in.	Output compared with an equal four-cycle engine
		Indicated	Brake				
400	333	9·0	7·8	86·7	80·7	70·0	1·55
600	500	14·0	11·8	84·3	83·7	70·5	1·57
800	667	19·15	15·7	82·1	85·8	70·4	1·56
1000	833	24·7	19·5	79·0	88·8	70·2	1·56
1200	1000	30·0	22·5	75·0	89·7	67·3	1·50
1400	1167	34·8	23·0	66·1	89·2	59·0	1·31

The extreme right-hand column shows the ratio of twice ηp to 90, this latter figure being taken as the value of ηp attained with a modern four-cycle engine of the same size. It will be noted that at all speeds excepting the highest the Dolphin engine gave rather exceeding 50 per cent. more power than would have been given by the corresponding four-cycle two-cylindered engine.

In fig. 458 a graph is given drawn from the table of results above; it will be seen that the brake horse-power attains a maximum value of 23 at the comparatively low piston speed of 1200 ft. per minute.

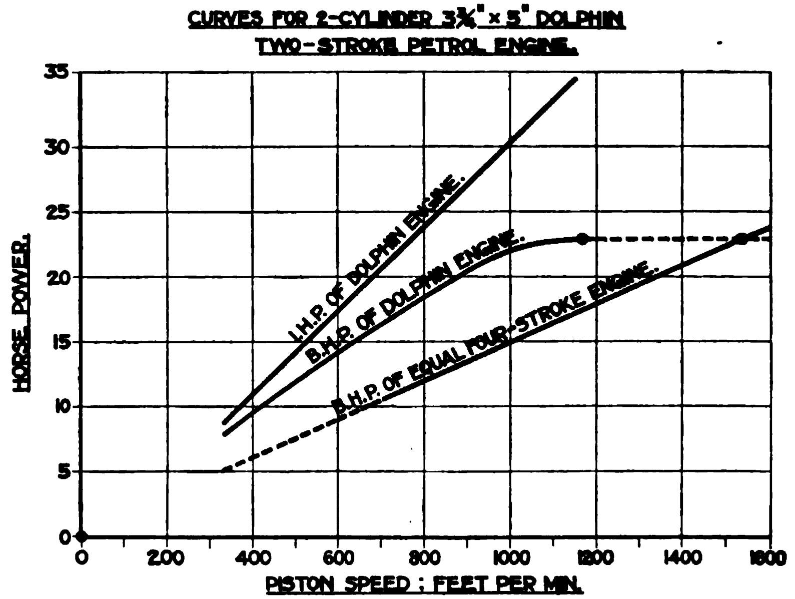


FIG. 458

The lowermost curve is that of brake horse-power of an equal four-stroke modern engine of good design having a constant value 90 for ηp from 700 to 1600 ft. per minute piston speed.

In cases where comparatively low revolution speed is a desideratum, as, e.g. in marine service, the advantage of this two-stroke engine is that roundly 50 per cent. more brake power is available than from an equal-sized four-stroke engine; but if high revolution speed be not an objection the two-stroke motor is soon at a disadvantage, as the brake power is seen to fall off rapidly above a piston speed of 1200 ft. per minute, while that of the four-stroke increases by a straight line law up to 1600 ft. per minute or more. In this case the four-stroke overtakes the two-stroke in brake power when its piston speed is raised to about 1550 ft. per minute, as shown on fig. 458. This case illustrates well the fundamental advantages and disadvantages of the small quick-speed, two-stroke petrol engine; at comparatively low speeds there is a gain in power output which, as has been seen, in a favourable case may amount to upwards of 50 per cent. in excess of that of an equal sized four-cycle engine; but if revolution speed be unrestricted, the very short time during which the ports of the two-cycle engine are open results at high speeds in imperfect scavenging, low volumetric efficiency, and consequent loss of power; accordingly at high revolution speeds the four-cycle engine is still to be preferred. The saving of weight, space, and cost, if any, resulting from adoption of the two-cycle engine is, in general, at present very small; its leading advantages are the more regular sequence of working impulses and the increased power output at comparatively low speeds as compared with an equal sized four-cycle engine.

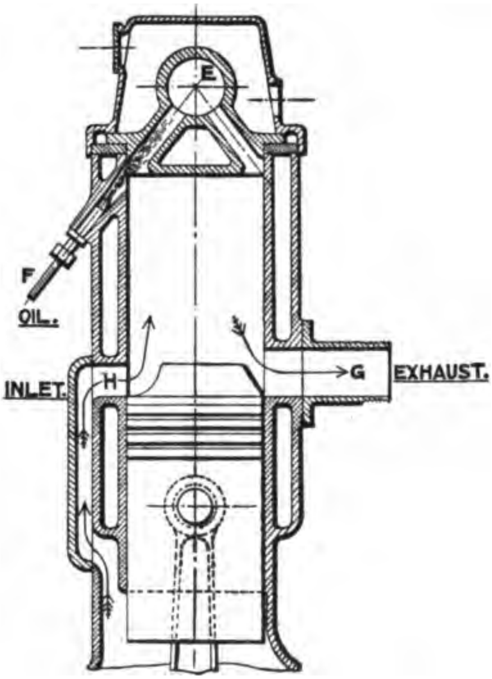


FIG. 459

An interesting modification of the two-port Day type of engine is found in the Swedish 'Bolinder' motor (Rundlöf's patent), which runs, however, on heavy oil, Scotch shale oil being frequently used; a diagrammatic section through the cylinder is shown in fig. 459. Air alone is drawn into the crank-chamber by the piston suction through leather flap valves on the sides of the crank case, and this air enters the cylinder under slight pressure near the end of the working stroke through the port H; the exhaust is simultaneously being discharged through the port G, and the air entering at H exerts the usual scavenging action, any slight loss through G being comparatively immaterial. The return of the piston compresses the air to a pressure of 120–130 lbs. per sq. in. into the combustion chamber, and slightly before the top of the stroke is reached the charge of fuel oil is injected into the hot bulb E formed in the cylinder head, the heat of which ignites the mixture as in the Hornsby engine; the oil is injected from the nozzle F and passes, with some of the compressed air, up one inclined passage to the bulb and down the other, after ignition, into the combustion chamber, thus thoroughly scavenging the bulb. To start the engine the bulb is blow-lamp heated for about a quarter of an hour; so soon as the engine is running normally the heated bulb causes the ignition to become entirely automatic.

The cylinder is supplied with its charges of oil by a small eccentric-driven plunger pump; water-spray may be injected into the combustion chamber when desired, to prevent knock at full load.

The engine speed is controlled by means of the fuel pump; between the driving eccentric and the horizontal pump plunger is a rocker and wedge-shaped striker, the thin end of which is horizontal and faces the pump plunger; the governor varies the position of the striker, and thus the stroke of the plunger, causing a corresponding variation in the quantity of charge injected into the hot bulb. The speed is also hand-controllable, the striker being arranged to be raised or lowered at will by a small hand-wheel, thus varying the charge as before.

The engine is easily and rapidly reversed by aid of an auxiliary fuel pump; the oil feed pump has two plungers, one placed horizontally, the other vertically; in normal running the horizontal plunger supplies the charge, the vertical being used only for reversing. To reverse, the engine is slowed and the vertical plunger thrown into action, the horizontal being then non-operative; this vertical plunger injects a small charge of oil into the cylinder during the compression stroke, causing an early ignition which checks the motion of the engine; after several such pre-ignitions the engine comes momentarily to rest and then commences to revolve in the opposite direction, when the vertical pump is cut out and the horizontal again brought into play. The pre-ignitions occur at low compression, with consequent absence of serious shock or jar to the engine during reversal.

Small slow-speed oil pumps are used to supply oil under pressure to the main bearings, big ends, &c.; the oil pumps are located above a small oil tank and driven by a shaft operated by ratchet gear from the crankshaft.

Single-cylindered engines are started—after a preliminary heating of the bulb—by turning the flywheel by hand in the reverse direction to that in which it is to run until a firing impulse occurs ; this does not necessitate, of course, more than a half turn. All engines with more than one cylinder are started by pressure from an air receiver in which air can be stored by the engine when running at a pressure of about 160 lbs. per sq. in. As in most oil engines, water spray injection is found very beneficial during long runs at full load ; about half as much water is generally used as fuel oil.

The 60 ft. × 19 ft. 3 ins. × 7 ft. 3 ins. fishing-boat 'Bolinders VII' was fitted in the after cabin with a two-cylinder Bolinder engine of 13 ins. bore and 13·4 ins. stroke ; this engine develops 80 horse-power at 325 revolutions per minute ; the piston speed is thus about 725 ft. per minute, while ηp has the low value of only 27½ lbs. per sq. in. ; the trawl-winch was driven by an independent single-cylinder Bolinder engine of 10 horse-power.

A recent noteworthy performance is that of the 65 ft. boat 'Lingueta,' fitted with a 30 horse-power Bolinder engine, which left Weymouth on December 28, 1911, and entered Pernambuco harbour on January 31, 1912 ; the voyage of 4500 miles was made at an average rate of 140 miles per day, the vessel having put in at Las Palmas only for stores.

Bolinder engines are constructed of the inverted vertical type with single cylinders up to 80 BHP ; with two cylinders to 160 BHP ; and with four cylinders to 500 BHP. A single-cylindered horizontal type is also made from 5 to 60 BHP, and fitted as a portable engine from 5 to 20 BHP.

In the two-stroke 'Reliable' motor of Webster & Bickerton crank-chamber compression was not used, but the bottom end of the cylinders was closed, and formed a pump and initial compressor for the mixture as in an early design by Mr. James Robson. A diagram is given in fig. 460 ; the carburetted air enters through a non-return valve A during the up-stroke of the piston, and fills the space B B B B ; during the downward (i.e. working) stroke this is compressed until the inlet port C is uncovered, when it streams into the upper part of the cylinder past the deflector D as usual. E is the exhaust port opened by the piston rather earlier than the inlet. Leakage from the crank-chamber and the addition of engine oil 'mist' to the carburetted air are thus avoided.

In order to diminish the height of the engine due to the presence of the piston rod, Messrs. Webster & Bickerton employed a very short connecting-rod, using a large crosshead guide with ample bearing surfaces. These engines were built of the open type with separate cylinders, having equal bore and stroke; the four-cylinder, 40 horse-power design had 7 in. cylinders with 7 in. stroke, and ran normally at about 400 revolutions per minute; this engine was produced for use in lighters, launches, and yachts.

The motor boats of the United States range in size from the small open 16 ft. craft with 2 horse-power single-cylinder motor to the triple-screw decked vessel of 100–120 ft. in length fitted with three sets of engines each developing about 200 horse-power.[1] These boats, including pleasure yachts, passenger launches, ferries, tugs, fishing-boats, oyster dredges, &c., are all run on petrol, notwithstanding the risk attending the use of this very inflammable fuel on shipboard, particularly in enclosed engine rooms.

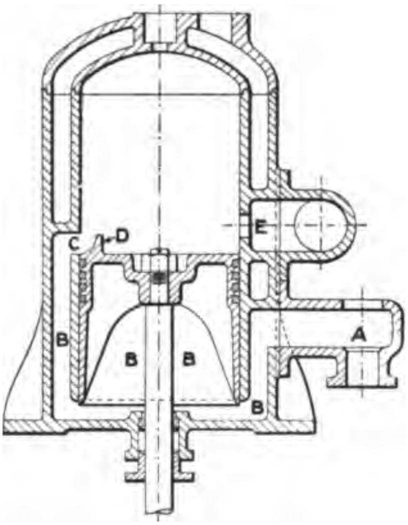


FIG. 460

An early large petrol marine set was the twin pair of six-cylinder 12 ins. × 14 ins. engines built by the Standard Motor Construction Company of America for the boat 'Gregory.' These were open reversible engines, the cylinders being supported on steel columns; the joint output was roundly 700 horse-power; starting and reversing were effected by compressed air.

The boat was only 70 ft. in length, with a beam of 11½ ft. and a 4 ft. draught; she succeeded in crossing the Atlantic under her own power, starting from New York on February 8, 1905; she was delayed at the Bermudas, through damage caused by stormy weather, until March 19, finally, after many minor mishaps, arriving at Algiers on May 17.

Another early example of a large marine petrol engine was the eight-cylinder, four-cycle, 7¾ ins. × 9 ins. engine developing 175 horse-power at 650 r.p.m. fitted by J. Craig of New York to the motor-boat 'Onontio' in 1904. The cylinders were separate and carried on light steel stanchions standing in a manganese bronze bed-plate; starting was effected by a ratchet and pawl-lever. The three-bladed propellor was 30 ins. in diameter and 60 ins. in pitch; the boat had a length of 60 ft., beam of 7 ft., and draught of 1 ft. 6 ins., and attained a speed on trial of nearly 28½ miles per hour in the Hudson river.

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[1] As, e.g. in Mr. Fleischmann's 'Whirlwind.'

In 1910 Mr. Craig constructed two four-cycle, six-cylinder, 11 ins. × 12 ins. engines, each giving 300 horse-power at 450 revolutions per minute, for installation in a 120 ft. steel boat. In these engines also the crank-chamber is open, the cylinders being carried on bright steel stayed columns as in the earlier 'Onontio' engine.

The pistons have cooling and strengthening ribs about ⅛ in. × 1 in. cast on the under side of their crowns; the connecting rods are of forged steel of ordinary marine pattern with the big end brasses lined with Babbitt metal; the crankshaft is 4 ins. in diameter.

The valves are operated by overhead rocking arms actuated by a camshaft carried along on the port side near the tops of the cylinders; in each cylinder head, in order to obtain the necessary area, there are two inlet and also two exhaust valves, each 3¼ ins. diameter with a lift of $\frac{9}{16}$ in.; each rocking arm operates two valves nearly simultaneously. One inlet is however caused to open very slightly before the other, partly to ease the load on the rocker and partly to cause the engine to run more quietly. The inlets open 1° late and close 6° late; the exhausts open at bottom stroke and close about 10° late; there is thus an overlapping timing of 9°. About one inch from the bottom of the stroke an auxiliary exhaust port is opened by the piston as in a two-cycle engine, and through this the greater part of the exhaust gas escapes, the overhead exhaust valves having only to pass out the remainder. All exhaust piping is water cooled, and the cooling is assisted by internal ribs in the piping.

The many disastrous fires and explosions that occurred on the earlier petrol-equipped boats brought vividly before engineers the necessity with this fuel of paying the most careful attention to the disposition of the tanks, piping, &c. While there seems no doubt but that on the whole kerosene is a safer fuel for marine purposes than petrol, it yet appears that when suitable precautions are taken the risk of fire with the lighter spirit can be made small. It must be remembered that not only kerosene but also petrol is regularly transported in bulk in specially constructed tank steamers with safety, although at first fires and explosions were not infrequent even on boats carrying bulk kerosene; an instructive account of some of these disasters is given by Sir B. Redwood ('Petroleum,' Vol. I, p. 505). The almost exclusive use of petrol in American motor-boat practice shows also that practical freedom from serious fire risk is attainable with proper arrangements. In Great Britain, partly on account of the greater cost of petrol, and partly from considerations of safety, the employment of petrol in large decked craft has, so far, been comparatively rare; a recent case, however, is that of Lord Montagu's 60-ft. 13-knot yacht 'Ytene,' fitted with twin four-cylinder, 75 horse-power, 6 ins. × 6 ins. Daimler petrol engines, usually run at 800 revolutions, and developing at this speed about 60 horse-power each. The petrol is stored in two 80-gallon brass tanks fitted in a special lead-lined compartment right aft; holes near the floor level drain off any liquid in the event of a leak, while holes in the deck covering maintain a constant air circulation in the space surrounding the tanks. All the copper piping from the tanks to the engines is carried *outside the hull* in a recess below the covering board until the engine room is reached, and coned pipe joints are used throughout.

The chance of any liquid petrol or vapour collecting in the vessel is thus small; the fuel supply is controlled at the tanks, but the supply to each engine can also be cut off by means of cocks at the filters on each carburettor branch. Some constructors have placed the petrol tanks and carburettors on deck, the induction piping alone entering the engine room. The Board of Trade requires a drip trough, draining overboard, to be fitted under each carburettor to prevent any leakage of petrol reaching the bilges.

Another recent large British petrol marine outfit is that of the motor yacht 'Cachalot,' engined by the Maudslay Motor Co. of Coventry in 1911. The boat is 50 ft. × 10 ft. × 3 ft. 6 ins., with a registered tonnage of 22. She is fitted with two six-cylinder, 5 ins. × 5 ins. engines, running right- and left-handedly respectively, at 900 revolutions per minute. Trials on the measured mile at Long Reach showed the boat to have a mean rate of 9·6 knots with the starboard engine alone, and 10·4 knots with both engines running.

A view of the engines is given in fig. 461; White & Poppe carburettors are used (*v*. Chap. IX) and centrifugal governors connected up to the carburettor throttles prevent the engines racing in heavy seas.

Forced lubrication is employed, the pumps being capable of supplying oil to the bearings at the high pressure of 150 lbs. per sq. in. when driving at full power against a head sea; large oil filters are fitted, and the oil is cooled by passing through pipes in the cold water sump.

The cooling water is supplied by large pumps of the gear-wheel type through a baffled intake, and delivered into a large filter-box or sump; surrounding this box are coils of copper piping through which the lubricating oil is cooled, as already mentioned. A branched delivery proceeds from each pump, one branch supplying the cylinder jackets, the other the water-cooled exhaust chamber at the side of the engine; from the cylinder jackets and exhaust jackets the water passes into a 'steam pot' or collector, shown clearly in fig. 461, placed above the engines, whence it is discharged overboard. The steam pots are fitted with relief cocks on top by which any entrapped air or steam in the circulating system may be at once liberated. A small two cylinder Coventy-Simplex engine is installed at the forward end of the engine room, and is used both for starting purposes and also for driving a lighting dynamo. The control rods are carried up through the deck to the wheel-house above the engine-room.

Very small flywheels are fitted, and no aluminium is used, the crank-chambers being steel castings fitted with large doors through which the main bearings may be adjusted and connecting-rods and pistons withdrawn when necessary. Overhead valves of the standard Maudslay type are used, and the overhead driving gear can be hinged back so as to give easy access to the valves at any time. Double ignition is employed, viz. high-tension magneto and coil and battery respectively.

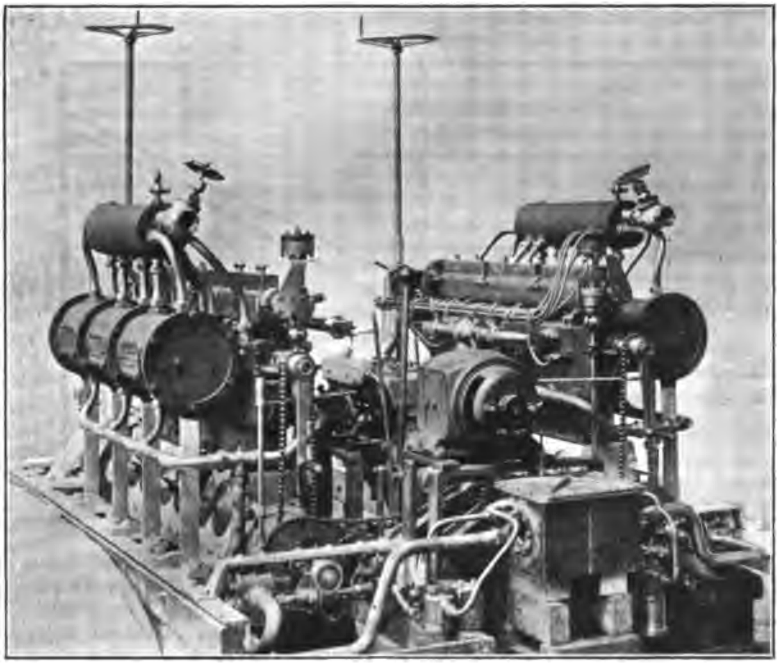

FIG. 461

A power-speed graph for these engines is given in fig. 462; it will be noted that, owing to rather better adjustments, the port engine gave somewhat higher power results than the starboard, though the engines were exactly similar in all details of construction. At 900 revolutions per minute an aggregate of 114 horse-power was developed; the tests extended to a speed of 1200 r.p.m., corresponding to a piston speed of 1000 ft. per minute, the total horse-power being then 148. The corresponding values of ηp are 85½ lbs. per. sq. in. at 900 r.p.m. and 82¾ lbs. per. sq. in. at 1200 r.p.m. The petrol consumption at full load is 0·66 pint of petrol per BHP hour, corresponding to a thermal efficiency of $\frac{2545}{18,600 \times 0{\cdot}66} = 0{\cdot}23$. These engines can, if desired, be worked on paraffin (though not so fitted in the 'Cachalot'); with this fuel about 0·7 pint per BHP hour is required, and the total power developed is only 85 to 90 per cent. of that obtained with petrol, as usual. Messrs. Maudslay do not recommend paraffin, considering that 0·760 petrol is but little higher in cost than paraffin (kerosene), and that there is no greater risk of fire when proper arrangements are made. Unfortunately it has often happened that the first indication of a petrol leak has been an explosion below due to diffusion of vapour from escaped petrol; by careful and continuous attention to the piping details, efficient ventilation of the engine room, and avoidance of any drainage of fuel into the bilges the danger is minimised. It may be mentioned that in the event of fire occurring chemical extinguishers of the 'New Era' and similar types are quite remarkably efficient; some interesting experiments with an apparatus of this kind on an old boat are illustrated and described in *The Motor Boat* of December 15, 1904.

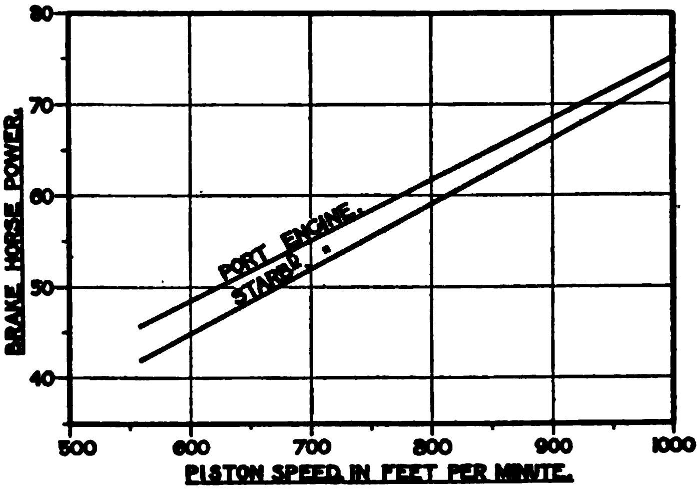

FIG. 462

Marine internal combustion motors, whether using petrol or kerosene, require to be of more substantial construction than the engines of motor cars, partly on account of the long periods during which they are required to run at full power and partly because full power must, in general, be developed at a lower piston speed than obtains in car engines, about 750 ft. per minute being the usual maximum in present practice for ordinary craft; moreover, when kerosene is used as fuel a lower compression must be adopted, and the output of power is only 85 to 90 per cent. of that obtained with petrol, necessitating again a heavier engine. When fitted in the boat it is impossible to get beneath the crankshaft, and the crank-chamber castings of marine motors are accordingly usually fitted with large doors which, on removal, permit the main bearings and big ends to be examined and adjusted, and in many cases the pistons to be removed also if desired, without disturbing the cylinders. On account of the action of sea water upon ordinary aluminium, crank-chambers are made of special alloys, cast iron, or steel; open engines are rare.

In the well-known small kerosene engines built by the Seal Motor Company of Hammersmith the engine is inverted, the cylinder being placed beneath the crank-chamber. The crankshaft, borne in phosphor bronze bearings, is closed in by a domed cover carrying the lubricators for the crank-pin and main bearings; removal of this cover permits examination of the piston, connecting-rod, and all bearings to be at once made. Owing to the combustion chamber being below the water line 'thermo-syphon' cooling may be employed.

In the smaller motor boats starting is effected by a geared-down handle and chain gear; in the larger sizes ratchet levers, or compressed air, or auxiliary starting motors are used. In fishing and commercial boats generally the engines should be of simple stout design either ring, chain, or wick lubricated, with splash lubrication in the crank-chamber, which should be an iron casting enclosing all the gear possible, including perhaps the thrust bearing; for the thrust balls should *not* be used; these engines are often called upon to run for weeks without any skilled attention. For better class boats forced lubrication is much to be preferred, in conjunction with an oil cooler.

The ignition of marine motors has frequently given a great deal of trouble where it has not been realised that the most perfect insulating arrangements are necessary on shipboard, and that practices which may suffice very well for cars are in many cases quite useless at sea. For boats in general high-tension magneto ignition is to be preferred, and this is very generally adopted in the larger types, though with big slow-running engines the low-tension magneto gives good results; the battery and coil system is sometimes fitted as a reserve; in the smallest boats, where cost is a prime consideration, the expense of the high-tension magneto precludes its use, and the coil and battery is much used. In his valuable paper on internal combustion engined boats[1] Mr. F. R. S. Bircham has dealt fully with the imperfections usual hitherto in the ignition installations on motor boats, and has made some recommendations deserving careful consideration. Mr. Bircham points out also that the exhaust silencing arrangements on motor boats have necessitated a good deal of attention. The best method, albeit somewhat costly, is to fit a water-jacketed exhaust pipe and silencer of copper; recently the Köerting ejector has been employed, using water pumped under pressure to discharge all the exhaust products below the water level; this involves a pump, and by suitably proportioning the details a negative pressure may be maintained in the exhaust pipe; this system is compact and involves little piping. With large boats a simple and effective device, also requiring but little pipe work, is the funnel silencer, but objection is often made to this on the score of appearance. In some cases a copper exhaust is used into which all the circulating water is discharged; this alone does not provide sufficient quietness for any but racing boats, and the addition of a silencer is necessary; it is then, however, frequently found that the silencer is rapidly corroded; there is also a risk of water being drawn back into the engine, causing valve corrosion and ignition troubles; to minimise this risk a non-return valve opening into the exhaust from the air is fitted close to the cylinders. In small low-priced boats plain pipes asbestos-lagged are often used; but these become very hot, steam when wetted, and cannot be regarded as satisfactory. Mr. Bircham advocates the water cooling of all the exhaust branches from the cylinders. A water intake of ample area, protected by a strainer, and with a forward-looking hood is necessary. The circulating pumps on large slow-running engines may be of the plunger type; with fast-running engines gear or centrifugal pumps are used; pet-cocks should be fitted at all points of the circulating system where there is any risk of accumulation of air.

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[1] *Proc. I.A.E.*, vol. iii.

The low volatility of kerosene (paraffin) and heavier oils has rendered its successful employment as a fuel for internal combustion engines a problem of much greater difficulty than when petrol or similar volatile liquids are used; at the present day, though extensively employed in stationary, agricultural, and marine motors, it is not practicable as a fuel for the engines of road automobiles, on account of the continual variation in the power output of the engine in this kind of service.

With kerosene and heavier oils some form of vaporiser must be employed; an account of these, together with a description of some typical cases, will be found in Chap. X.

The complete four-cylinder 6 ins. × 8 ins. Thornycroft marine paraffin engine is illustrated in fig. 463; the vaporiser and its adjuncts are clearly shown.[1] This type is much used in fishing craft; with paraffin the engine develops 47 brake horse-power at 700 revolutions per minute on a consumption not exceeding 0·9 pint of oil per BHP hour; if petrol be used, the output of power increases to 53 BHP; as fitted in 80 ft. × 20 ft. × 8 ft. 6 ins. boats of 60 tons displacement a mean speed of 8 miles per hour is obtained. Water spray injection into the cylinders is employed, enabling higher compression pressures to be used, and producing smooth running at full load. Among recent large marine paraffin engine installations may also be mentioned the 700 HP twin sets supplied by Messrs. Thornycroft in 1908 for the propulsion of submarine vessels of the Royal Italian Navy. These boats are 98 ft. 6 ins. in length by 14 ft. 6 ins. beam, with a displacement of 150 tons; the radius of action at 10 knots is 500 miles. Each engine comprises eight cylinders of 12 ins. bore; the stroke is 8 ins., and normal speed 550 revolutions per minute; restricted head room necessitated the adoption of a stroke-bore ratio less than unity. The eight-cylinder engine is obtained by coupling together two standard four-cylinder units, one of which is illustrated in section in fig. 464; the engines are coupled tandem, the shafts being connected with their crank-planes mutually at right-angles, thus giving a regular sequence of working impulses with a period of one quarter of a revolution; the order of firing is 17384625; the engines work on the four-stroke cycle.

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[1] See Fig. 390 for a sectional view.

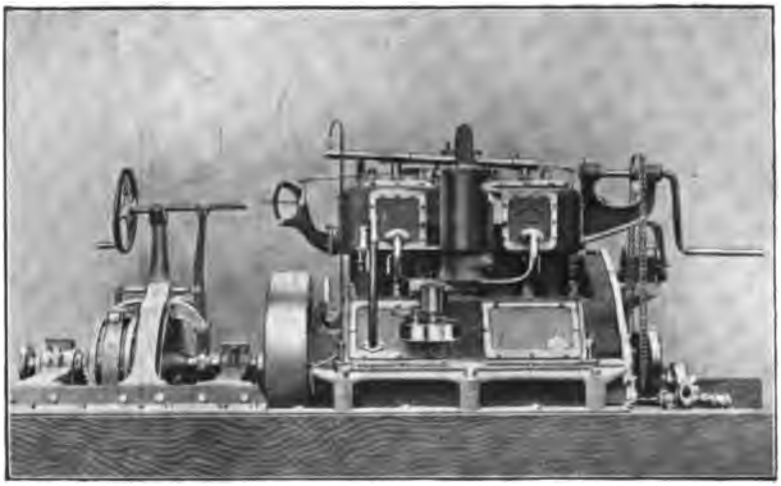

FIG. 463

Overhead valves driven by an elevated camshaft give easy access to the crank-chambers, in which large covered openings are provided through which the main bearings may be adjusted and pistons and

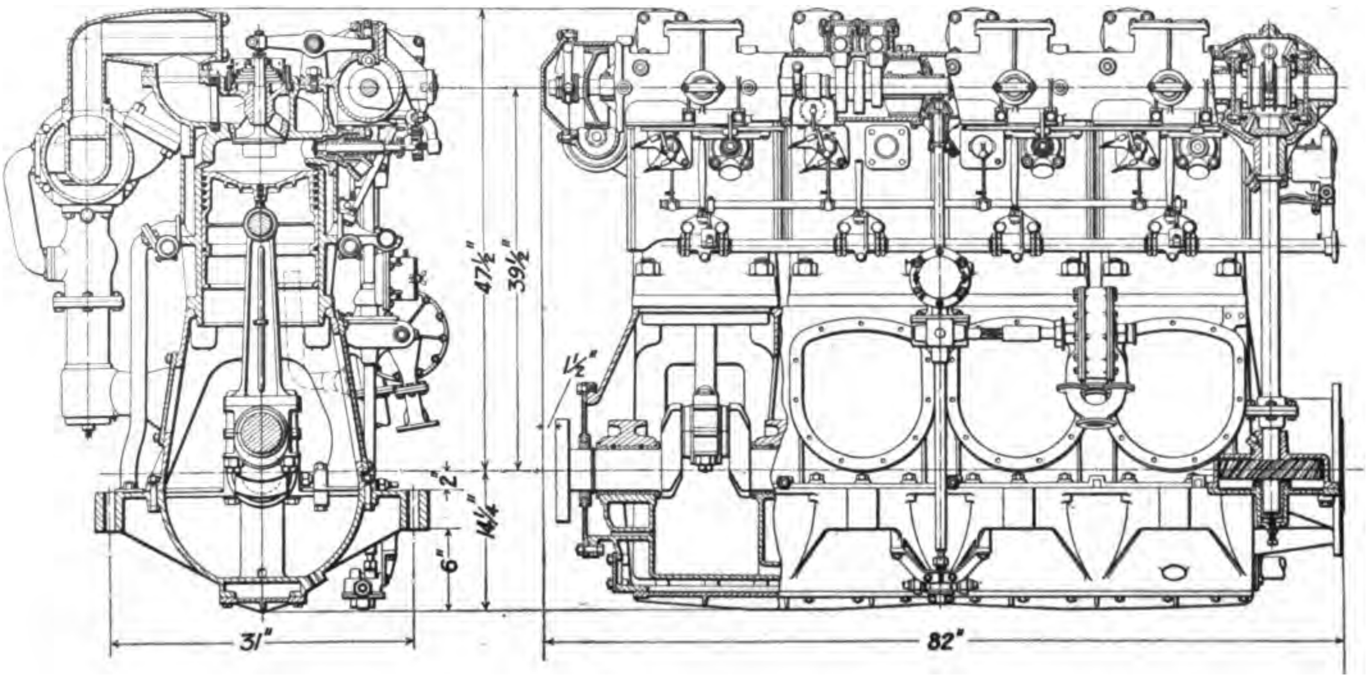

FIG. 464

connecting-rods withdrawn, without lifting the engine; the upper half of the crank-chamber is of aluminium and the lower of bronze.

The water circulating pumps are independently driven by electric motors, thus enabling the engines to be rapidly cooled down after coming to rest in the event of a breakdown. The inlet valves are self-contained in their own cages, but the exhaust valve seatings are formed in the cylinder heads to facilitate cooling, these latter valves being inserted and removed through the inlet valve orifices in the cylinders.

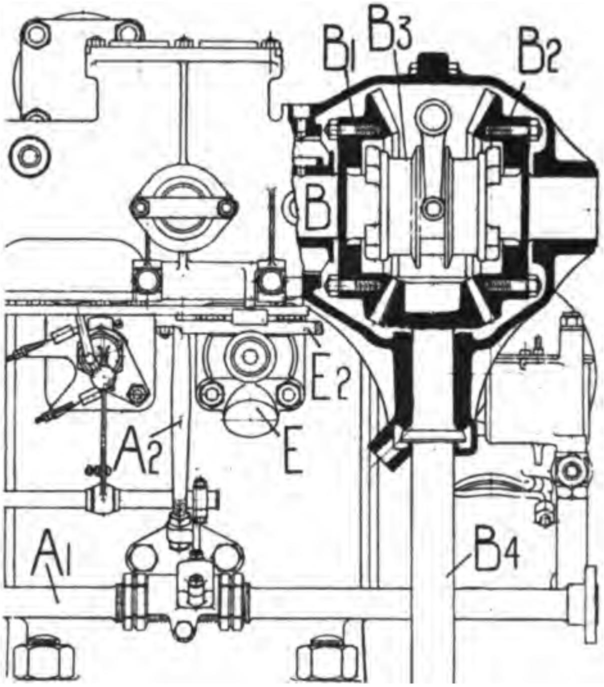

FIG. 465

Reversing is effected by aid of the device illustrated in fig. 465; on the camshaft B are fitted two equal, free, opposed bevel wheels, B_1 and B_2; by means of the dog-clutch B_3 either of these may at will be connected with the shaft B. At the end of the vertical driving shaft B_4 is a bevel pinion always in mesh with both B_1 and B_2. In order to reverse, the dog-clutch is moved across from B_1 to B_2 just before the engine comes to rest; the reversing impulse is then given to the pistons by compressed air admitted early through the air starting valves; so soon as the reverse motion is established the normal firing cycle is taken up. It is noteworthy that the camshaft revolves always in the same direction with this arrangement, and further that the device is limited to engines having opposed cranks; it was for this reason that the engine was made up of two tandem four-cylinder units, each unit having cranks at 180°.

Starting was by compressed air admitted through special valves already mentioned. The paraffin vaporisers fitted were substantially to the design as illustrated in Chap. X, fig. 390, enabling either kerosene or petrol to be used; a water drip was fitted to each cylinder to diminish the explosion knock when at full load. Ignition was by low-tension magneto, the firing current being served to the several cylinders by a low-tension distributor, thus avoiding the difficulty with the usual 'omnibus-bar' arrangement that the 'earthing' of one igniting plug affects all the others.

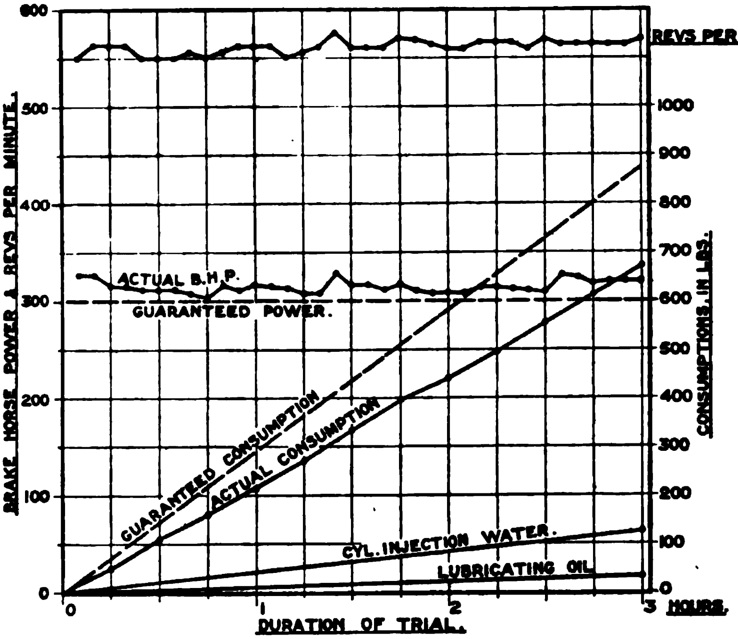

FIG. 466

Lubrication was forced; oil contained in the crank-chamber sump was delivered by a gear-pump to the main bearings and thence through crank ducts to the big ends; from these, a pipe along the connecting-rod supplied the gudgeon bearings. The exude from the big ends lubricated the cylinder walls by whirling off from the cranks; the camshafts and valve rockers were lubricated by gravity from sight-feed oil cups.

The crank-chambers were lubricated and kept cool and free from inflammable gases which might collect from piston leakage, by means of an electrically driven exhausting fan, fresh air being drawn in through gauze-covered orifices in the inspection doors.

A full-load official trial was made in February 1908. Before the commencement of this trial the engine was run for some time so as to become thoroughly warmed throughout, about 3¾ gallons of kerosene being thus used. The official full-load trial was of three hours' duration, and the fuel used was 'Phœbus' paraffin having a specific gravity of 0·820. The total consumption during the trial amounted to 83½ gallons, corresponding to roundly 0·71 pint per BHP hour. During the trial 13 gallons of water were admitted to the cylinders as 'drip feed,' corresponding to 0·11 pint per BHP hour. A diagram showing some of the results of this trial is given in fig. 466. It will be seen that the speed varied only between 550 and 575 revolutions per minute, the average being 560·7 during the whole period; the BHP also ranged from 305 to 327, with a mean value of 314·7. Thus the average piston speed throughout the trial was 747·4 ft. per minute, and the average value of ηp roundly 61½ lbs. per sq. in. The compression pressure employed, in order to avoid pre-ignition, was only about 55 lbs. per sq. in., the volume ratio of compression being 3·18. With petrol as fuel a BHP of 450 was obtained; thus the power developed with paraffin was in this case only about 73 per cent. of that with petrol. The consumption of lubricating oil during the trial amounted to 10¾ pints per hour; the amount of cooling water was not actually measured, but was estimated at 160 gallons per minute; the temperature on entering the false bottom of the crank-chamber was 59° F., and on leaving the silencer-jacket 95° F., the total rise of temperature being thus 36° F.

The inlet and exhaust valves had each a diameter of 5 ins., with a $\frac{13}{16}$ in. lift; the weight of one piston and connecting-rod, all complete, was 151 lbs.; the connecting-rod was 22 ins. long between centres. The weight of one eight-cylinder engine complete was, roundly, 7½ tons; taking the mean of 314·7 BHP with paraffin as fuel, this corresponds to 53½ lbs. per brake horse-power only.

The Gardner is a well-known design in marine practice. In this engine an oil pump is not now employed, the supply of paraffin to the mixing device being either gravity or pressure fed, according to the location of the oil reservoir. The general arrangement of the mixer is diagrammatically shown in fig. 467; K is the kerosene supply branch; the oil enters the small duct D through the hand-regulated needle valve

N; A is the air inlet. V is a light spring-supported disc carried on a spindle terminating at the top in a needle valve P, which shuts off the kerosene supply when the engine is at rest; C communicates with the lamp-heated vaporiser.

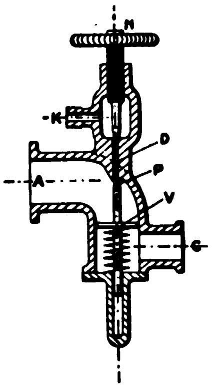

FIG. 467

On each suction stroke of the engine the disc V descends, opening the kerosene supply at P and permitting a mixture of air and spray to pass through C into the vaporiser and thence past the inlet valve into the cylinder.

Additional air is admitted to the mixture through a group of two or sometimes three small automatic mushroom valves in the cylinder head, through which also enters the water spray used in all the Gardner paraffin engines; when the throttle, or governor-controlled inlet valve, is closed these valves open more fully and allow a charge of fresh air to enter the cylinder during each suction stroke.

Governing is effected by varying the lift of the inlet valve by means of a centrifugal crankshaft governor.

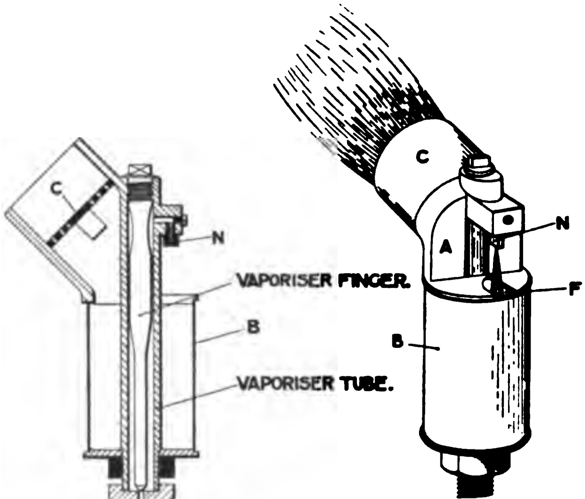

FIG. 468

Each cylinder has its own mixer, vaporiser, and blow-lamp, unlike the type just previously described, where one lampless vaporiser suffices for any number of cylinders; with the lamp-heated vaporiser there is no risk of the engine stopping after running for some time at light load, and full power can be attained quickly after such running.

The Gardner paraffin blow-lamp, or 'pressure-feed burner,' is illustrated in fig. 468; its mode of operation is similar to that of the ordinary Swedish, 'Etna,' or hand blow-lamp in which a jet of kerosene vapour is mixed with air and burned, giving a Bunsen flame.

In Messrs. Gardners' design the kerosene under a pressure of 2 or 3 lbs. per sq. in. enters the burner through the hand-regulated needle valve K, and passing up the hot central tube is vaporised and issues in a jet directed downwards from the small nipple N through the orifice F in the cover of the 'burner tube' B, inducing a stream of air to enter with it; the mixed air and vapour pass through the burner grid C and burn at the upper surface of this, giving a blue cylinder of flame by which the vaporiser is kept hot. The burner is started in the usual manner by heating the upper part A with a hand blow-lamp; about once weekly it must be taken to pieces and thoroughly cleaned, the 'vaporiser finger' being also removed and freed from deposit. A yellow or smoky flame indicates either too large a hole in the nipple N, or too cold a burner, the oil not being completely vaporised when it issues from the nipple, or lastly, the holes in the burner grid may be obstructed. Ignition in the Gardner marine engine is by low-tension magneto.

Fig. 469 shows a longitudinal and transverse section of the four-cylinder, four-cycle 6½ ins.7 × ½ ins. Gardner paraffin marine engine, showing the location of the blow-lamps, mixing chamber, and vaporisers; it will be noted that the vaporising chamber is merely the space below and around the inlet valve in this design. The inlet and exhaust valves are on opposite sides of the cylinders; the two air inlet poppet valves will be noted in each cylinder head with overhead water drip piping and regulating cocks. On the extreme left the low-tension magneto and an ignition plug are shown. Large inspection doors give access to the crank-chamber for executing bearing adjustments, &c.

This engine develops 48 BHP at 600 revolutions per minute, the corresponding piston speed being 750 ft. per minute and brake mean effective pressure, ηp, about 64 lbs. per sq. in. The extreme length is 6 ft.; breadth 3 ft. 4 ins.; and height 3 ft. 9 ins. The total weight is roundly 2600 lbs., a weight per BHP of about 54 lbs. As fuel, kerosene of 0·825 sp. gr. and flash point by Abel close test as high as 200° F may be used; it is important with this, as in other oil engines, that the fuel should have no tendency to form tarry deposits.

Engines up to 75 BHP are started by hand by aid of the usual chain and free-wheel device; compression relief cocks are fitted. Above 75 BHP compressed air self-starters are used, the compressed air being stored in a reservoir and maintained by a separate small air compressor.

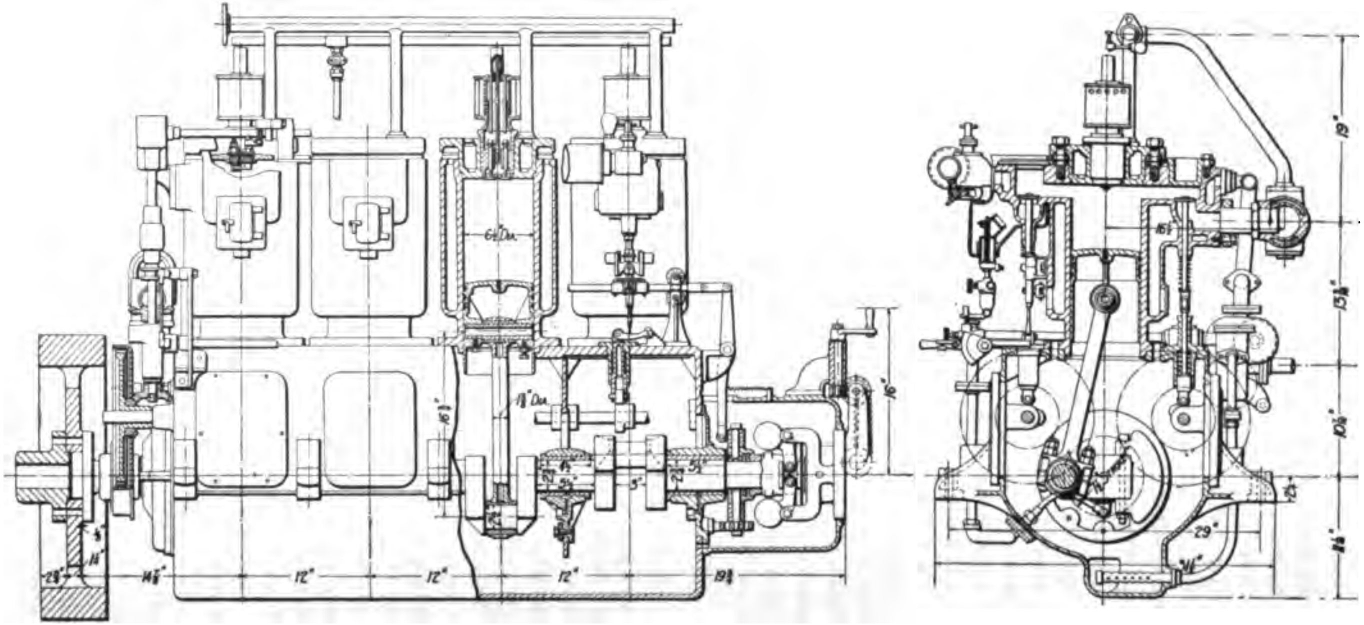

FIG. 469

The Gardner four-cycle paraffin marine engines are built with 1, 2, 3, 4, 6, and 8 cylinders, and for all powers from 5 to 200 BHP.

Messrs. Blackstone & Co. have recently introduced a four-cycle crude oil engine for marine purposes, of which an external view is shown in fig. 470; this illustrates the four-cylinder, 9½ ins. × 10 ins., 80 horse-power design; it is single acting, and runs normally at 450 revolutions per minute, corresponding to a piston speed of 750 ft. per minute. During the suction stroke air alone enters the cylinder and is compressed on the following stroke to a pressure of about 150 lbs. per sq. in.; the corresponding temperature is insufficient to ignite the heavy oils employed, and ignition is accordingly accomplished by means of a hot bulb on the cylinder end into which a small jet of oil of constant amount is sprayed by means of air supplied from reservoirs at a pressure of 400 lbs. per sq. in.; the heat of the bulb ignites this spray; and a flash of flame is projected into the combustion chamber, in which a second jet of oil is similarly sprayed, as in the Diesel, during the first portion of the working stroke; the flash ignites this mixture,

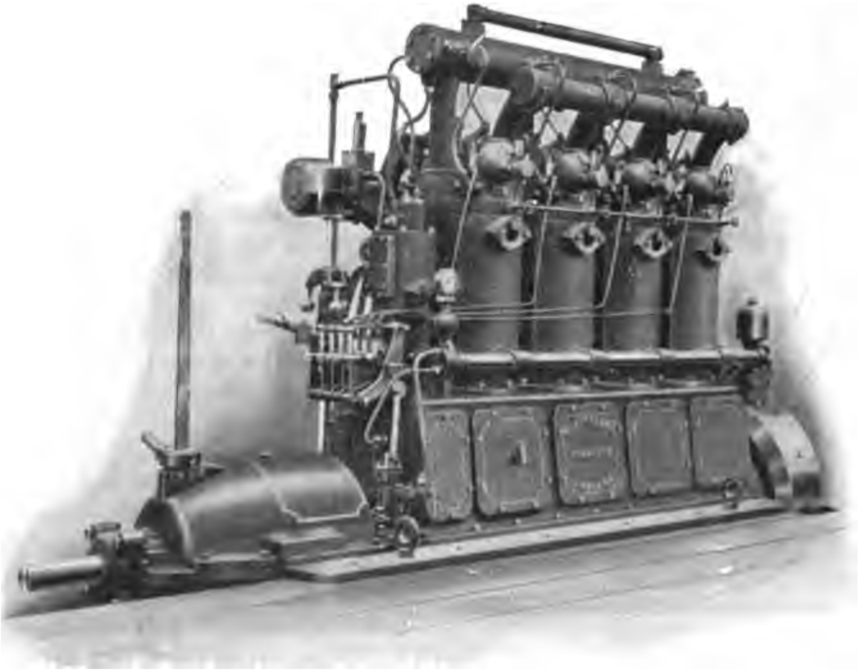

FIG. 470

which burns at approximately constant pressure during the admission period; characteristic indicator diagrams are exhibited in fig. 471. The constant volume pressure rise is due to the bulb spray, the constant pressure line to the main spray; the maximum pressure attained at full load is only about twice the compression pressure, i.e. some 300 lbs. per sq. in.; in the full load diagram of fig. 471 the ratio of maximum to mean effective pressure is 3·68 to 1 only. The engine is of the enclosed type with separately cast cylinders having detachable heads; the cylinders and heads are, of course, water-cooled. The crank-chamber is of cast iron, with large inspection doors on each side; forced lubrication is provided for the main bearings. A diagrammatic sectional view is given in fig. 472; from which it will be seen that the piston has a false crown and four spring rings, and that the connecting-rod is of the ordinary marine type with adjustable gudgeon bearing. A is the hot-bulb, the main oil spray entering at the centre of the cylinder head B; the flash from A passes down through the oblique passage shown and ignites the contents of the combustion chamber. C and D are the inlet and exhaust valves, also in the cylinder head; at E compressed air may be admitted for starting and reversing purposes.

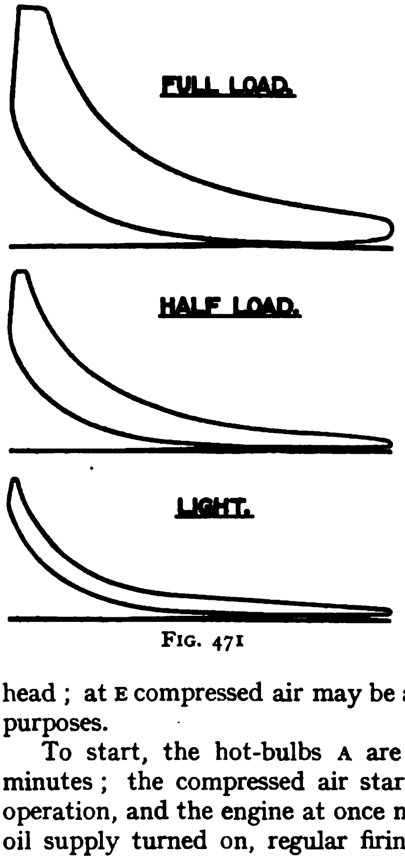

FIG. 471

To start, the hot-bulbs A are blow-lamp heated for about ten minutes; the compressed air starting valves are then brought into operation, and the engine at once moves off; the air is cut off and the oil supply turned on, regular firing at once occurring; after a few minutes the blow-lamps may be extinguished, ignition then continuing automatically. All the valves are actuated by a single camshaft on which are two sets of cams for both inlet and exhaust valves, either set being brought into action at will by a sliding motion along the shaft. In order to reverse, the oil supply is momentarily cut off, when the engine at once slows down; during slowing down, the cams are slid along the camshaft so as to bring the reverse set into operation; just before the engine comes to rest the air starting valve is opened early, giving the engine an impulse in the opposite direction. After a few such revolutions the air is turned off and the fuel oil supply simultaneously turned on, when the engine at once takes up the working cycle, running in the reverse direction; with the arrangement adopted, the inlet valves become the exhaust valves and *vice versâ* when running astern. The water-cooled air compressor supplying the reservoir with air at 400 lbs. per sq. in. is fitted in the after end of the crank case, and driven directly from the main shaft. The fuel oil is delivered from a tank to the sprayers by a governor-controlled pump; the oil first reaches the hot-bulb sprayer, providing a small charge, of constant amount, for ignition; the excess passes on to the main sprayer and is admitted to the combustion chamber during the first part of the working stroke. The governor varies the stroke of the oil pumps and thus proportions the main spray to the load requirements of the engine. The engine will run at from 160 to 450 revolutions per minute.

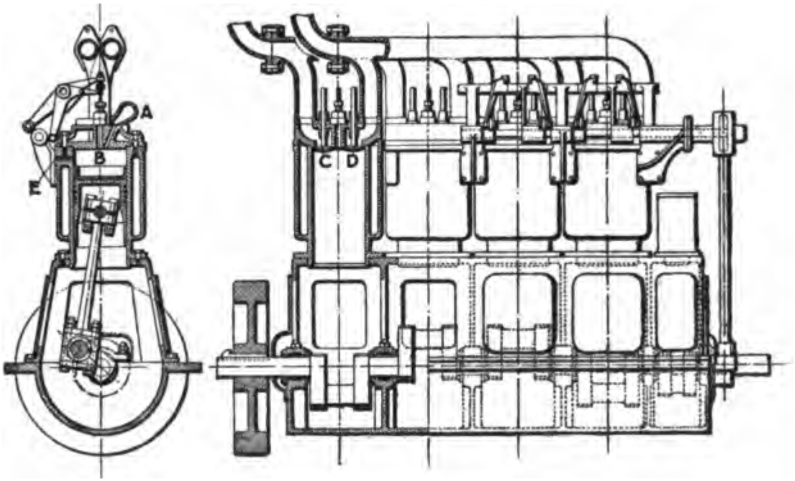

FIG. 472

The Blackstone crude oil engine is now running in a number of vessels with satisfactory results; for example, in the power barge 'Rose,' owned by Messrs. Leetham of York, a four-cylinder, 80 horsepower engine as described is installed; when loaded, and towing also a loaded lighter, the 72-mile trip from York to Hull is made in about 11 hours, giving a mean speed of roundly 6½ miles per hour; the fuel oil consumed on the run is roundly 55 gallons, and lubricating oil 2 gallons.

The Griffin Engineering Company of Bath have also recently produced heavy oil engines designed for marine use; a general view of one of the four-cylinder type is given in fig. 473; they are built with one, two, or four cylinders and in sizes from 15 to 350 horsepower; the makers guarantee that they may be run on shale oil, Scotch fuel oil, Texan liquid fuel, &c., and even on turpentine refuse and creosote.

The reversing gear, shown in fig. 473, and the thrust block are carried in a stiff cast-steel frame bolted to the crank-chamber base; the crankshaft terminates in a bevel wheel, a similar wheel being fitted to the foward end of the tail shaft. Meshing with these is a pair of epicyclic bevel pinions carried on a common spindle borne in the outer casing; at either end of this casing is a friction clutch. When the friction clutch adjacent to the engine is let in the bevel system is locked, and the drive from engine to propellor is direct; when the friction clutch at the after end is let in, the casing carrying the epicyclic pinions is fixed, this clutch connecting to a stationary drum forming part of the casing; the pinions then revolve on their stationary spindle, and the propellor shaft rotates in the opposite direction to that of the crankshaft. When both clutches are freed the epicyclic bevels revolve, the casing turning idly at one-half the crankshaft speed, and the propellor shaft stands still, thus giving the 'free engine' position. This type of reverse gear is satisfactory for small and moderate powers; but there are practical difficulties in employing it with very large engines.

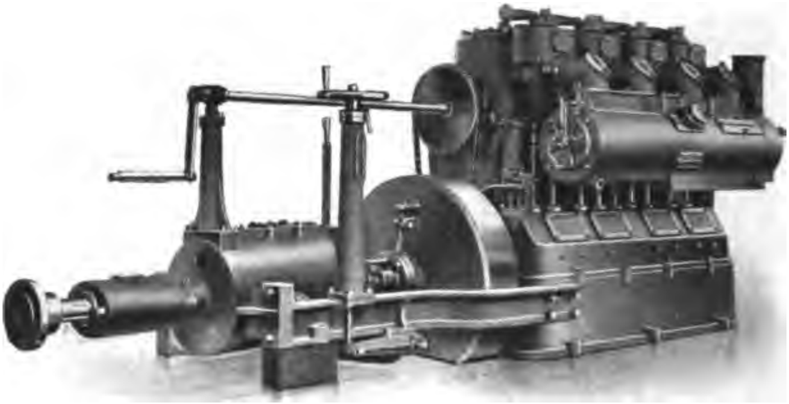

FIG. 473

The Griffin marine engine is of the enclosed four-cycle, single-acting type; the 75 horse-power design has four 10½ ins. cylinders, 10½ ins. stroke, and runs normally at 350 revolutions per minute.

The specially distinguishing feature of the Griffin engine is the large cylindrical horizontal vaporiser, shown clearly in fig. 473, extending the whole length of the engine. This consists essentially of a large exhaust-heated pipe, in the centre of the after end of which is fitted a nozzle through which the fuel oil is sprayed by air at a pressure of about 20 lbs. per sq. in.; in the case of very viscous oils a preliminary heating takes place in order to increase the fluidity. The spray cloud entering the heated cylinder is at once vaporised, any tarry products collecting at the bottom and continuously draining away during the running of the engine; the temperature of the vaporiser may be regulated to suit the fuel used and power output of the engine. The governor acts directly on the oil duct to the nozzle, thus varying the quantity of the fuel sprayed; the consumption is stated to average about 0·7 lb. of oil per BHP hour. At each end of the vaporiser is an adjustable air inlet; at the spraying end the main air is admitted.

A short elbow connects each cylinder with the vaporiser; the mixture thus reaches the cylinders in a very hot condition, and, due in part to this thermodynamically objectionable practice and in part to its nature, the very low compression pressure of only 45 lbs. per sq. in. is adopted; with kerosene engines in general, owing to risk of pre-ignition, compression pressures of 55 to 70 lbs. per sq. in. only are usual. The air for spraying the oil is supplied by a small air pump driven from the after end of the crankshaft; to start the engine the air pressure is first raised by hand pumping, and the air regulators in the vaporiser are then opened; the oil being turned on, the spray cloud is next ignited in the vaporiser, causing a flame filling its entire volume and quickly raising it to the necessary temperature; the use of a separate blow-lamp is thus rendered unnecessary. When sufficiently hot the air doors in the vaporiser ends are closed, thus extinguishing the flame, and on turning the engine over it at once starts running; there is an additional automatic air valve on the vaporiser admitting extra air to the mixture, and this is in constant operation during the running of the engine.

The larger engines are turned at starting by an unusual method: the flywheel is connected with the crankshaft by a friction clutch; this clutch is released, and by means of a multiplying-up chain gear the flywheel is caused to revolve rapidly by hand; the clutch is then let in, when the momentum of the wheel carries the engine round and thus starts it off.

The Diesel engine has for some years been tentatively applied to marine service, and much attention is now being concentrated upon it as one of the most promising means of solving the problem of marine propulsion on the large scale by means of the internal combustion engine.

On account of the more regular sequence of working impulses, and the simpler reversing arrangements involved, the single-acting, two-stroke type of engine is considered by most engineers as the type likely to be principally adopted for this purpose in the near future; owing, however, to diminished temperature troubles, better scavenging action, and better engine balance, several well-known builders, including Messrs. Burmeister & Wain, The Netherlands Company, and Messrs. Barclay, Curle, & Company, adhere at present to the single-acting, four-stroke type. The Diesel engines of the Nürnberg Company are of the two-stroke, double-acting type; with this type the presence of the piston-rod and stuffing-box introduces the practical difficulty that the fuel oil cannot be injected from the centre of the cylinder cover, and recourse must be had to the use of two or three injection valves, involving increased cost and complication. With double-acting engines, also, water- or oil-cooled pistons become a necessity, and this is a practice which has not so far found much favour with English engineers. Since 1903 Messrs. Nobel Bros. of St. Petersburg have run the 1100-ton, 7½-knot boat 'Vandal' on the Caspian Sea; this vessel has three propellers, each driven by a separate Diesel engine; the propellers are reversed by an electrical device. Nobel's 1100-ton, 8-knot boat 'Sarmac' is propelled by two four-cylinder Diesel engines, each of 180 horse-power, at 200 revolutions per minute.

The Mirrlees Diesel Company has for some years past supplied enclosed type four-cycle high-speed Diesel engines of 45 to 300 BHP to H.M. Navy as auxiliaries; these engines run at 400 revolutions per minute, and are direct coupled to dynamos; they use the same fuel oil as is used for firing the main boilers.

A 60-foot launch named the 'Dreadnought' was fitted by the Diesel Engine Company of London in 1909 with a four-cylinder 7·1 ins. × 9·85 ins. Sulzer-Diesel two-stroke engine, and experimental runs were made on the Thames until the latter part of 1911. The authors were present at a demonstration run made in September 1911; using Texan fuel oil (sp. gr. 0·925), the engine developed about 85 BHP at 345 revolutions per minute; the piston speed was thus 567 ft. per minute, and brake mean effective pressure 62·8 lbs. per sq. in.; the mechanical efficiency of the engine is stated to be roundly 70 per cent. The blast air pressure employed was 850 lbs per sq. in., compression pressure 500 lbs. per sq. in., and outlet temperature of cylinder jacket water 160° F. The engine could be slowed down to about 140 revolutions per minute, in which case the oil supply and air blast of two of the four cylinders were rendered non-operative, these cylinders then pumping idly. A sectional view and description of this engine will be found in Chap. X (fig. 447).

Throughout, the running was quiet and steady; from full speed ahead the engine was reversed to full speed astern with ease and certainty well within five seconds; the methods of starting and reversing are described in detail in Chap. X.

The first ocean-going passenger vessel propelled by Diesel engines was the East Asiatic Company's 12-knot boat 'Selandia,' gross registered tonnage 4964, designed to carry a dead-weight cargo of about 7400 tons. The 'Selandia' is 370 ft. between perpendiculars, 53 ft. beam, and has a moulded depth of 30 ft.; the passenger accommodation is in a deck house amidships. The vessel plies between Copenhagen and Bangkok. There are twin screws, each driven at 140 revolutions per minute by an eight-cylindered, single-acting, four-cycle Diesel engine having cylinders of 20⅞ ins. bore and a stroke of 28¾ ins. At full speed the gross indicated power is 2500; from indicator diagrams taken during the first voyage to London in February 1912 it was found that a mean effective pressure of 91 lbs. per sq. in. was obtained at 129 revolutions per minute, corresponding to a piston speed of 620 ft. per minute, an IHP per cylinder of 146, and a gross IHP for the sixteen cylinders of 2336. The engines are of the crosshead design, but the crank-chambers are enclosed; the crosshead type promises to become general, and it is also probable that open crank-pits will become standard practice. The 'Selandia' has no funnel, the exhaust gases being carried away up the mizen mast.

The engines are started and reversed by compressed air at 300 lbs. per sq. in. Two four-cylinder auxiliary oil engines are fitted, each developing 250 IHP at 230 revolutions per minute; each of these engines drives a generator furnishing current for working other auxiliary machinery, and also operates a two-stage air compressor which stores air at 300 lbs. per sq. in. in four large reservoirs; air at this pressure is used for starting and reversing. From these reservoirs at 300 lbs. pressure the high-pressure compressors, driven from the forward end of each main crankshaft, deliver air for the oil fuel injection blast at pressures up to 1000 lbs. per sq. in. into three steel bottles for each main engine, one of which is the working bottle, the remaining two being spares. Each compressor is alone able, in emergency, to supply both the main engines; moreover, the exhaust valve of one each of the eight engine cylinders may be readily replaced by a non-return valve, so that, by turning off the oil fuel to this cylinder, it may be operated as an air pump and used to replenish the 300 lb. pressure reservoirs when desired. The 'Selandia' also carries a donkey boiler giving steam for heating the vessel, but in case of need the steam can also be used to drive a small Reavell compressor by which the air reservoirs may be filled up. The whistle is driven by air-pressure from the 300 lb. reservoirs. The donkey boiler is oil-fired, the same fuel being used as in the main engines.

At the top of each main engine casing is a daily supply fuel oil tank of capacity sufficient for 12 hours' normal working; the fuel oil is delivered to these by compressed air pumps; any water suspended in the oil has thus a better chance of separating out. From these daily supply tanks the oil flows by gravity to the force-feed pumps.

In the spring of 1912 the largest marine Diesel set under construction was by Messrs. Krupp for a twin-screw oil-carrying vessel of 10,800 gross tonnage. Each main engine comprises six two-stroke single-acting cylinders of 22·45 ins. bore and 39·4 ins. stroke; the indicated horse-power per cylinder is nearly 300, the gross IHP for the twelve cylinders being 3500.

The largest IHP per cylinder so far (1912) being built under commercial conditions for marine service is thus, roundly, 300; in Germany, however, an experimental three-cylinder, two-stroke, double-acting Diesel engine of 6000 IHP, i.e. 2000 IHP per cylinder, was constructed at Nuremberg in 1911, with which many tests were made. Early in 1912 a serious accident occurred, apparently through a leaky fuel oil valve causing a pre-ignition of the charge, resulting in a cylinder burst, accompanied, unfortunately, with loss of several lives.

The table[1] on p. 793 gives some leading particulars of the principal Diesel engines for marine service built, or in process of building, in the early part of 1912.

Discussing the marine applications of the Diesel engine, Mr. J. T. Milton, M.Inst.C.E., said in 1911:

'In a steam engine many casualties may occur in which one cylinder only is disabled, and the engine can still be worked by the remaining two without a serious loss of power or speed, each of the remaining pistons getting a higher pressure upon them than when working under normal conditions. In the Diesel engine, under similar conditions, one cylinder may be disabled, and the others will still be able to work at their full pressure; the loss of work then will be less in proportion to the number of cylinders employed.

'In the four-stroke cycle engines actually made, or in process of building, for large ships, either six or eight cylinders are being used per shaft. In the two-stroke cycle single-acting engines four cylinders seem to be preferred, whilst the double-acting engines being built have three cylinders. Both five and six cylinders have been proposed for the two-stroke cycle.

'In a steam vessel, besides the main propelling machinery, there are a number of auxiliaries, all worked by steam. It remains to be

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[1] Compiled from *Engineering*, March 8, 1912.

Some Large Diesel Marine Engines built, or building in 1912

Gross tonnage of vessel	Number of propellers	Gross IHP of engines	Total number of cylinders	Diameter of cylinders in inches	Stroke in inches	Type of engine	Remarks
4920	2	1800	8	17·72	22·05	Two-stroke, single-acting Carels	
3500	1	1700	6	18·875	25·5	Two-stroke, double-acting Nürnberg	
4964	2	2500	16	20·87	28·74	Four-stroke, single-acting B. & W.[1]	Asiatic Co.'s 'Selandia.'
2200	1	1100	6	22·05	39·4	Four-stroke, single-acting 'Spoorweg'	
4500	2	2300	12	18·7	31·5	Two-stroke, single-acting Krupp	
10,800	2	3500	12	22·45	39·4	Two-stroke, single-acting Krupp	
2640	1	—	4	18·1	32·28	Two-stroke, single-acting Carels	
8500	2 or 3	6000	24	24·8	31·5	Four-stroke, single-acting B. & W.	Projected only, in March 1912
3380	2	—	16	19·69	25·98	Four-stroke, single-acting B. & W.	
—	1	1000	6	17·72	21·27	Two-stroke, single-acting type	

[1] B. & W. = Burmeister & Wain.

considered what would be the best way to provide for doing their work in a vessel fitted with Diesel propelling machinery.

'The more important are the following: (1) Steering gear, (2) whistle, (3) donkey pumps for bilge and fire service, (4) electric light machinery, (5) distillers, (6) steam heating apparatus, (7) water ballast pump, (8) winches, (9) windlass, (10) ventilating apparatus in passenger vessels. Of these, the first six and the last are always required at sea, the remainder are generally only needed when the vessel is either in or near port. Probably for these latter the best solution is to continue the present practice of working them by steam from a donkey boiler, which may be fired with oil fuel. This boiler may be put into action only when in or about to enter port. For the remainder there are some alternatives. The electric light machinery presents no difficulty, as it can be worked by a small oil engine of either Diesel or other type, using the same kind of fuel as the main engine, resulting in a similar economy of fuel as will be obtained with the main engine. This engine might also be used for working the pumps for bilge and fire purposes, as these pumps are only required on emergencies when part of the electric light may be dispensed with. The ventilating arrangement can be electrically driven, the motive power being the same engine as is used for electric light. The ballast pump may be a steam pump worked from the donkey boiler, but it may be dealt with by working it by means of a small separate oil engine.

'Besides these auxiliaries the Diesel engine requires some of its own. The quantity of circulating water required to keep the cylinder cool is very large. This is probably best supplied by a pump worked from the main engines, but in view of a possible derangement of this pump it will be desirable to have also a separate connection with the ballast pump. The air compressors for supplying the compressed air used for injecting the fuel will also be worked by the main engine, but a supplementary compressor will be necessary to keep up the supply of air which is needed in manœuvring. This must be worked by a separate engine.

'Further, in the event of the receivers losing the air pressure from any cause, it is necessary to have another small engine which may be started without compressed air (say, a small paraffin engine), in order to obtain a sufficiency of compressed air, preferably in a small separate reservoir, to start the auxiliary compressor engine.

'In some designs of two-stroke Diesel engines each cylinder has been arranged with its separate scavenge pump. In other designs one large double-acting pump supplies all the cylinders. In the former arrangement, if one cylinder fails from any cause the remainder are not interfered with, and if a scavenge pump valve gives out, only the one cylinder which the pump serves will be disabled. Where, however, one large pump serves all the cylinders, while it is recognised that the arrangement of the pump with one piston and one mechanically worked valve is very simple and most unlikely to get out of order, yet it must be admitted that its failure, if it does occur, will occasion a complete stoppage of the engine.

'Possibly in recognition of this, some recent designs provide for two independent scavenge pumps, both requiring to be in use under normal working conditions. It is improbable that both can be damaged at one time, and seeing that the air supply in each cylinder contains three times the quantity of oxygen chemically necessary for the complete combustion of the fuel, it is expected that in the event of one pump only being at work there will be sufficient air supplied to either work all the cylinders with a reduced fuel supply, or to work, say, three out of four of the cylinders at nearly full power.

'Much more consideration will have to be given to the accessibility of every working part than has been done in some designs. Every part of the engine which requires periodical inspection or occasional adjustment must be made easily accessible.

'Provision must be made for the possibility of replacing the crankshaft without lifting the engines out of the ship.

'In short, the Diesel marine engine should be Diesel only as regards the cylinders and their accessories, and of the ordinary marine type in all other respects.'

A sectional view of Messrs. Beardmore's 'semi-Diesel' engine is given in fig. 474; this is a two-stroke, two-port engine with crank-chamber compression and hot-bulb ignition, much as in the 'Bolinder' engine already described earlier in this chapter. On the up-stroke of the piston air enters the crank-chamber through automatic flap valves, and is compressed slightly therein on the down-stroke until the upper edge of the piston overruns the port B, when the air enters the cylinder from the crank-case and assists in expelling the exhaust from the previous working stroke through the port C. On the ascent of the piston the air entrapped in the cylinder is compressed into the combustion-chamber and hot-bulb D to a pressure of about 150 lbs. per sq. in., and at the instant of maximum compression the charge of oil is injected into this by an air blast at a pressure of 400–500 lbs. per sq. in. through the needle injection valve E shown. A preliminary blow-lamp heating of the hot-bulb is, of course, necessary before starting; when the engine is under way the bulb temperature is maintained by the heat of the successive explosions, and the ignition then becomes automatic. Using kerosene as fuel, the maximum pressure during combustion is only about 300 lbs. per sq. in., while with Texan or Solar oils the maximum is reduced to about 200 lbs. per sq. in.

The engine illustrated had four cylinders of 9 ins. bore with a stroke of 13 ins., with a two-stage air compressor driven by a fifth crank at the forward end; one of these engines was fitted early in 1912 to Lord Graham's 85-ft. yacht 'Mairi' for experimental purposes. The normal full engine speed was 350 revolutions per minute, corresponding to a piston speed of 760 ft. per minute, and propelling the yacht at a rate of 10¼ knots. An indicator diagram from this engine is reproduced in fig. 475, from which it will be noted that at 300 r.p.m. the maximum pressure of explosion was 275 lbs. per sq. in., and mean effective pressure 52½ lbs. per sq. in., corresponding to 32·9 indicated horsepower per cylinder, or a total of 131·6 for the four cylinders.

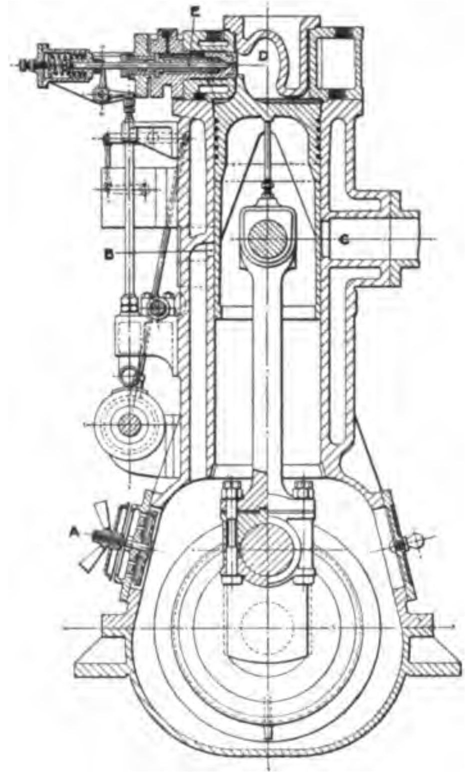

FIG. 474

The oil consumption was 0·56 pint of Texas fuel oil per BHP hour, corresponding to a brake thermal efficiency of about 21·8 per cent.

In the Deutz Co.'s four-cycle crude oil 'Brons' engine, also, air alone is compressed to a pressure of about 400 lbs. per sq. in., and the charge of fuel oil is injected suddenly at the instant of maximum compression by a device enabling the highly compressed air blast of the Diesel engine to be dispensed with. There is no constant pressure combustion period in the cycle of this engine, and the sudden injection of the fuel charge causes a maximum explosion pressure of about 750 lbs. per sq. in. to be attained; the Brons engine is accordingly somewhat costly and massive in construction, and about 30 horsepower per cylinder is the present limit of size.

The two-cylinder, 24 HP Brons engine has a bore of 7⅞ ins. and stroke of 9½ ins., and runs normally at 340 revolutions per minute;

this corresponds to the somewhat low piston speed of 540 ft. per minute, and mean effective pressure of about 60 lbs. per sq. in. The ratio of mean to maximum pressure is thus about 1 : 12½. Including the flywheel, the weight of the 24 HP engine is 272 lbs. per horse-power; starting is by compressed air; ignition is automatically effected jointly by the compression and an internal perforated 'ignition cap' fitted to the inner end of the fuel valve guide, and projecting into the combustion chamber, the heat of this being maintained by the successive explosions during the running of the engine.

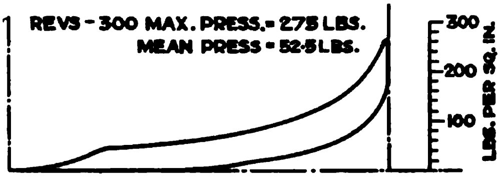

FIG. 475

The Fiat Co. of Turin have also recently produced a series of enclosed, inverted vertical, two-stroke, heavy-oil marine engines; the 500 horse-power design includes six cylinders, and this engine runs normally at 500 revolutions per minute.

Messrs. Thornycroft have recently (No. 21,585 of 1911) patented, for cases of marine propulsion, the combination of an installation of steam turbines capable of developing the maximum power required, and the corresponding steam generating apparatus, with an internal combustion engine plant of sufficient power to propel the vessel at low speeds, together with means of enabling the turbines and internal combustion plant to be respectively connected up at will with the propeller-shafting.

Taking, for example, the case of a torpedo boat destroyer requiring 15,000 indicated horse-power at full speed, only about 1000 effective horse-power suffices for the ordinary cruising speed of about 15 knots. In such a case turbines of an aggregate indicated horse-power of 15,000 would be installed together with, say, two 500 BHP Diesel engines in the usual case of a twin-screw boat.

When running at cruising speed the Diesels would be coupled up each to its own propeller shaft, and the main steam supply to the turbines cut off; any auxiliaries necessary to keep running could exhaust into the turbines, and the air pumps being kept in operation, a vacuum would be maintained in the turbines which would diminish their resistance to rotation. The boilers, or such of them as were not required, could be kept under banked fires, or if oil-fired could be shut down; in the case of oil-fired boilers the same oil would serve as fuel for the Diesel engines.

A number of possible arrangements are described, one of which is illustrated in figs. 476 and 477, showing in elevation and plan respectively two propeller shafts A provided each with a turbine B placed aft of oil engines C, clutch-coupled or geared to the shafts A. When cruising, the clutch or gearing of each oil engine is in action, these engines then driving the propellers; the turbines may then, if desired, be connected with the condensers Q.

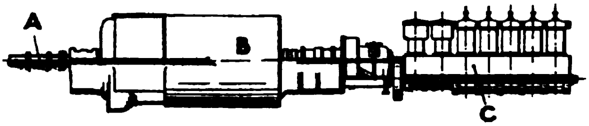

FIG. 476

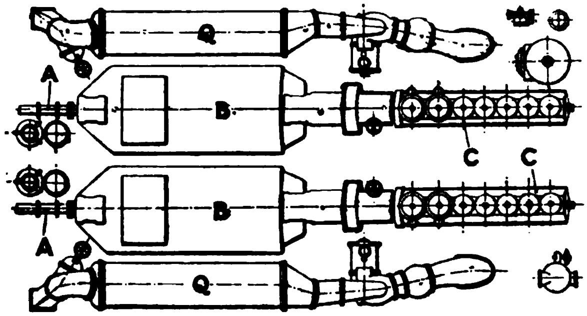

FIG. 477

When the turbines are to be used the clutches or gears are disengaged and the oil engines are thrown out of operation.

LLOYD'S REGULATIONS FOR MARINE INTERNAL COMBUSTION ENGINES

The rules adopted by Lloyd's for the survey of internal combustion engines running on petrol, paraffin, or heavy oils in marine service, are given hereunder:

General

Section 1. In vessels propelled by internal combustion engines, the rules as regards machinery will be the same as those relating to steam engines so far as regards the testing of material used in their construction and the fitting of sea connections, discharge pipes, shafting, stern tubes and propellers.

Construction

Section 2.—1. The following points should be observed in connection with the design of the engines.

2. The shaft bearings, connecting-rod brasses, the valve gear, the inlet and exhaust valves must be easily accessible.

3. The reversing gear and clutch must be strongly constructed and easily accessible for examination and adjustment.

4. In engines of above 60 BHP which are not reversible and which are manœuvred by clutch, a governor or other arrangement must be fitted to prevent racing of the engine when declutched.

5. Efficient positive means of lubrication (preferably sight feed) must be fitted to each part requiring continuous lubrication.

6. If the engines are of the closed-in type, they must be so fitted that the contained lubricating oil can be drained when necessary, and in wood vessels an easily drained metal or metal-lined tray must be fitted to prevent leakage of either fuel oil or of lubricating oil from saturating the wood work.

7. Carburettors, where petrol is used, and vaporisers, where paraffin is used, should be so designed that when the engine is stopped the fuel supply is automatically shut off. If an overflow is provided in the carburettor or vaporiser, a gauze covered tray with means of draining it must be fitted to prevent the fuel from flowing into the bilges.

Strong metallic gauze diaphragms should be fitted either between the carburettor (or vaporiser) and cylinders or at the air inlets.

8. If the ignition is electric, either by magneto or by coil and accumulator, all electric leads must be well insulated and suitably protected from mechanical injury. The leads should be kept remote from petrol pipes, and should not be placed where they may be brought into contact with oil.

The commutator must be enclosed; and the sparking coils must not be placed where they can be exposed to explosive vapours.

9. No exposed spark gap should be fitted.

10. In paraffin and heavy oil engines where lamps are used for ignition or for vaporising, these lamps should be fixed by some suitable bracket, and the flame enclosed when in use.

11. The circulating pump sea suction is to have a cock or valve on the vessel's skin placed on the turn of the bilge in an easily accessible position, and the circulating pipe is to be provided with an efficient strainer inside the vessel. The discharge overboard is to be fitted with a cock or valve on the vessel's skin if it is situated under or near the load line of the vessel.

12. A bilge pump worked by the engines or an independent power-driven bilge pump is to be fitted, to draw from each part of the vessel. In open launches this bilge pump may be omitted provided suitable hand pumps are fitted.

13. The cylinders are to be tested by hydraulic pressure to twice

the working pressure to which they will be subjected; the water jackets of the cylinders to 50 lbs. per sq. in., and the exhaust pipes and silencer to 100 lbs. per sq. in.

14. The exhaust pipes and silencer should be efficiently water cooled or lagged to prevent damage by heat, and if the exhaust is led overboard near the water-line, means must be arranged to prevent water from being syphoned back to the engine.

15. The machinery must be tried under full working conditions, the report stating the approximate speed of vessel, the number of revolutions of the engines at full power, both ahead and astern, and the lowest number of revolutions of the engines which can be maintained for manœuvring purposes.

Rules for Determining Sizes of Shafts

Section 3. The crank, intermediate, and other shafts if of ordinary mild steel are to be of not less diameters than as given in the following table. When special steel is used, the sizes are to be submitted for consideration.

1. For petrol or paraffin engines for smooth water services:

$$\text{Diameter of crankshaft in inches} = c\sqrt[3]{D^2S},$$

where D = diameter of cylinder in inches, S = stroke of piston in inches.

Four-stroke cycle	Two-stroke cycle	Bearing between each crank	Two cranks between the bearings
For 1, 2, 3, or 4 cyls.	1 or 2 cyls.	C = 0·34	C = 0·38
,, 6 ,,	3 ,,	C = 0·36	C = 0·40
,, 8 ,,	3 ,,	C = 0·38	C = 0·425
,, 12 ,,	6 ,,	C = 0·44	C = 0·49

For open sea service add 0·02 to c.

Diameter of intermediate and screw shafts in inches =

$$c\sqrt[3]{D^2S\,(n+3)},$$

where D = diameter of cylinder in inches, S = stroke of piston in inches, n = number of cylinders.

For smooth water services	For open sea services
c = 0·155 for intermediate shafts.	c = 0·165
c = 0·170 for screw shafts fitted with continuous liners.	c = 0·180
c = 0·180 for screw shafts fitted with separate liners or with no liners.	c = 0·190

In engines of two-stroke cycle, n is to be taken as twice the number of cylinders.

2. When ordinary deep thrust collars are used the diameter of the shaft between the collars is to be at least $\frac{21}{20}$ths of that of the intermediate shaft.

3. In the cases of Diesel and other engines in which very high initial pressures are employed, particulars should be submitted for special consideration.

Fuel Tanks and Connections

Section 4.—1. Separate fuel tanks are to be tested, with all fittings, to a head of at least 15 feet of water. If pressure feed tanks are employed, they are to be tested to twice the working pressure which will come on them, but at least to a head of 15 feet of water. If the tanks are made of iron or steel they should be galvanised.

2. Strong and readily removable metallic gauze diaphragms should be fitted at all openings on petrol tanks.

3. Paraffin or heavy oil tanks, not used under pressure, are to be fitted with air pipes leading above deck. Pressure-feed tanks and tanks containing petrol should be provided with escape valves discharging into pipes leading to the atmosphere above deck. The upper ends of all air pipes are to be turned down, and pipes above 1 inch diameter are to be provided with gauze diaphragms at the end.

4. No glass gauges are to be fitted to fuel tanks containing either petrol, paraffin, or heavy oil.

5. Filling pipes are to be carried through the deck so that the gas displaced from the tanks has free escape to the atmosphere.

6. Separate fuel tanks should be provided with metal-lined trays to prevent any possible leakage from them flowing into the bilges, or saturating woodwork. Arrangements are to be provided for emptying the tanks and draining the trays beneath them. For petrol tanks the trays must have drains leading overboard where possible or they should be gauze covered trays with means for draining them.

7. All fuel pipes are to be of annealed seamless copper with flexible bends. Their joints are to be conical, metal to metal. A cock or valve is to be fitted at each end of the pipe conveying the fuel from the tank to the carburettor or vaporiser. The fuel pipes should be led in positions where they are protected from mechanical injury and can be exposed to view throughout their whole length.

8. The engine room, and the compartment in which the fuel tanks are situated, are to be efficiently ventilated.

9. An approved fire extinguishing apparatus must be supplied.

Periodical Surveys

Section 5.—1. The machinery is to be submitted to survey annually. At these surveys the cylinders, pistons, connecting-rods, crank and other shafts, inlet and exhaust valves and gear, clutches, reversing gear, propeller, sea connections, and pumps are to be examined. The electric ignition is to be examined and the electric leads tested. The fuel tanks and all connections are to be examined, and, if deemed necessary by the Surveyor, to be tested to the same pressure as when new. If practicable, the engines should be tested under working conditions.

2. The screw shaft is to be drawn at intervals of not more than two years.

By order of the Committee.

16*th June*, 1910.

MARINE PRODUCER GAS PLANTS

Producer gas first introduced by Mr. Dowson in 1878 (*v.* Vol. I, p. 32) is now very largely employed in land installations, and has been applied to a small extent for the propulsion of marine gas engines.

So far, for this latter purpose, the suction producer has been employed, the gas being generated from anthracite coal, the variations in the composition of which may be inferred from the following results of analyses:

PERCENTAGE COMPOSITION OF ANTHRACITE COALS

No.	I.	II.	III.	IV.	V.
Carbon	75·0	81·0	86·4	89·0	91·5
Volatile matter	6·0	6·0	4·3	4·5	6·9
Sulphur	2·0	0·5	—	0·5	0·6
Water	2·0	3·8	3·4	2·0	—
Ash	15·0	8·7	5·9	4·0	1·0
	100·0	100·0	100·0	100·0	100·0

No. 1 is a poor anthracite; No. II is 'Coalite,' which, at a suitable cost, is a satisfactory fuel for this purpose; No. III is an American anthracite; No. V is Welsh anthracite. These are all characterised by their high carbon content and small proportion of volatile constituents, sulphur, and ash.

The power gas is obtained by passing a mixture of air and steam through incandescent anthracite or coke contained in a vessel termed a 'producer,' of which a great many types exist; further details are given in Chap. V of this volume. It will be sufficient here to say that the gas obtained consists roughly by volume of 17 per cent. hydrogen, 26 per cent. carbonic oxide, the remainder, comprising about 57 per cent. of the whole, being non-inflammable gas, mainly nitrogen; owing to the large proportion of carbonic oxide the gas is extremely poisonous. The calorific value of good producer gas of this kind is from about 140 to 160 B.Th.U. per cub. ft. at 32° F. and normal barometric pressure; as good coal gas has a heat value of about 640 B.Th.U. per cub. ft., it is seen that this producer gas has, at best, only one-fourth the calorific value of coal gas. The volume of air necessary for complete combustion is about one-tenth greater than that of the producer gas burned; these figures will enable a general idea of the nature of this gaseous fuel to be formed. As used in the gas engine, a *contraction* of about 7 per cent. occurs after combustion, which is rather larger than when coal gas is used.

Tests with a 9½ ins. × 18 ins. Crossley gas engine at 160 r.p.m. (*a*) with Manchester coal gas, and (*b*) with Dowson gas, showed that the gas used per HP hour was 19·4 cub. ft. in the (*a*) case, and 81·8 cub. ft. in the (*b*) test, figures which are substantially in inverse proportion to the respective calorific values of the two fuels. Tests of land installations of 'Premier' gas engines using Dowson gas of 155 B.Th.U. per cub. ft. heat value have shown a consumption of roundly 1 lb. of anthracite per BHP hour.

On account of the large nitrogen dilution, in addition to the presence of a portion of the exhaust gases from the preceding stroke, higher compressions can be used with power gas than with coal gas; special care must be taken to thoroughly mix the gas and air during admission to the cylinder.

An illustration of a recent simple anthracite suction producer plant, suitable for marine use, built by Messrs. Gardner, is given in fig. 478; this is designed for use with engines up to 180 BHP.

The generator, on the right, is constructed of mild steel plates with riveted seams, the lower part being lined with brick or cement to form an ashpit. Above the ashpit the separate cast-iron firebars are borne in a cast-iron frame riveted to the shell; at the level of the top of the fire-doors a second cast-iron frame supports the firebrick lining of the shell in the part containing the incandescent anthracite; the fire-bricks are separated from the steel shell by a packing of foundry sand. On the generator is fixed a cast-iron boiler through which a tube connects the fuel hopper on top with the fire; the interior of the boiler is always in free communication with the atmosphere at the main air inlet, so that the boiler is never subjected to any pressure;

large air-tight cast-iron doors on the generator render cleaning and clinkering easy.

In the pipe conducting the steam-saturated air to the ash-pit is fitted a hand fan for blowing up the fire when starting ; also a throttle

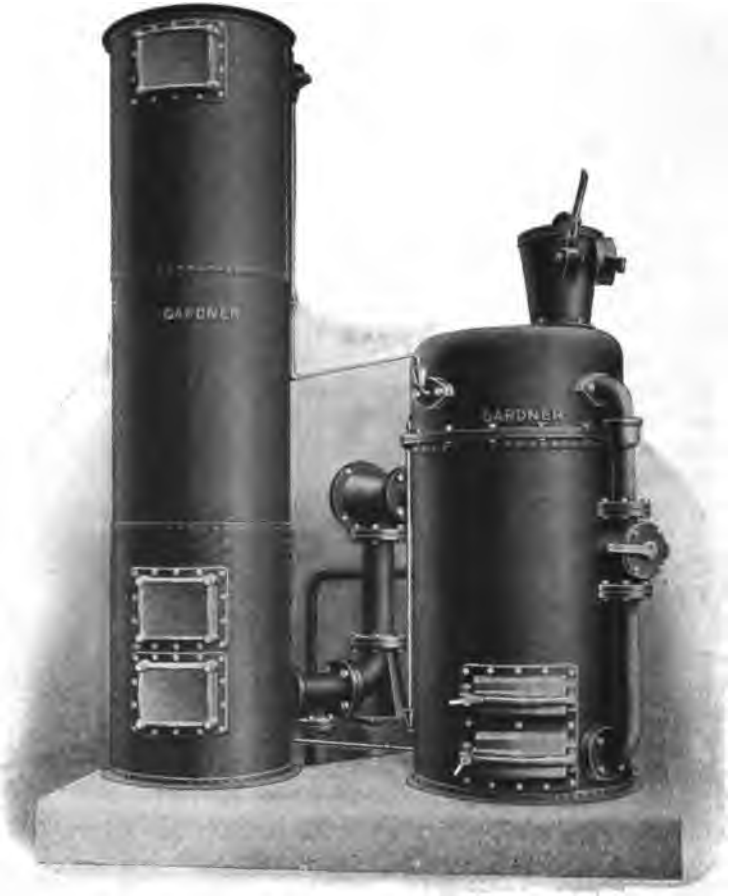

FIG. 478

for regulating the air supply. The steam-saturated air passes from the air-tight ashpit between the fire bars and through the incandescent anthracite into the upper portion of the generator ; the gas thus produced is hot and contains dust, tarry products, and other impurities ; it is accordingly next passed into the left-hand vessel termed a ' scrubber,' consisting of a cylinder of mild steel plates with riveted joints ;

the lower part of the scrubber forms a sump, above which is fixed a grid supporting a filling of coke; at the top is fitted a water sprayer. The hot gas drawn through the scrubber by the engine suction is cooled and purified and then passes on to the engine cylinders; the sprayed water collects in the sump and thence drains off into a 'water seal' box. In the pipe connection between the generator and scrubber is fitted a three-way cock by which the generator may be placed in communication either with the scrubber only or with the chimney only; the scrubber and generator cannot be both open to the chimney at the same time. The use of steam enriches the heat value of the resulting gas, cools the fire bars, and prevents an excessive temperature in the fire; great care is taken to make all joints air-tight throughout on account of the poisonous qualities of the gas.

For a 60 horse-power engine the total weight of the producer and plant is about 2¾ tons; the generator is about 3 ft. 3 ins. in diameter and 6 ft. 6 ins. in height; the scrubber about 3 ft. 0 ins. in diameter and 8 ft. 3 ins. in height.

In 1910 the sailing boat 'Castell San Nicolan,' plying between Palma and Barcelona, was fitted with an auxiliary 60 horse-power Gardner suction plant of this type; the 3-cylinder, 8 ins. × 9 ins. engine runs at 500 r.p.m. and a speed of 5 miles per hour is obtained at 45 BHP; the consumption of anthracite is stated to be about 6 cwts. for a run of 100 miles.

The first English boat to be fitted with an engine driven by suction producer gas was the 60 ft. × 10 ft. × 4 ft. launch 'Emil Capitaine'; an average speed of ten miles an hour was maintained on a 10-hour trial in 1905, the consumption of anthracite being stated as 412 lbs.

The four-cylinder, single-acting, four-cycle, 8¼ ins. × 11 ins. engine of this boat developed about 60 BHP at 300 revolutions per minute; reversal was effected by a Thornycroft reversing gear interposed between the engine and propeller shaft. The boat was built and engined by Messrs. Thornycroft at Chiswick, under the Capitaine patents.

The producer was placed in a separate well-ventilated compartment bulk-headed off from the engine room, but communicating with it through doorways; sufficient fuel was contained in the producer for a 12-hour run.

Messrs. Thornycroft also fitted a suction gas plant to the barge 'Duchess,' 71 ft. 6ins. × 7 ft. 1 in. × 3 ft. 6ins. draught, with 20 tons of cargo on board; the engine was of 30 horse-power. The installation of suction gas power effected a saving of three feet in the engine room length, and about four tons in the weight of the machinery. The 'Duchess' made an extended tour of some 600 miles through the English canals in order to test the plant in this type of service.

Herr Capitaine had already built a suction gas driven tug at Frankfort on the Maine, and in 1904 this vessel, named 'Gas Tug No. 1,' 44 ft. 3 ins. × 10 ft. 6 ins., fitted with a four-cylinder, 70 horse-power Capitaine engine and suction producer, was tested against the steam tug 'Elfrieda,' 47 ft. 0 ins. × 12 ft. 0 ins., fitted with a 75 horse-power triple-expansion steam engine. A run from Kiel to Hamburg and back was made by these two vessels in very stormy weather, at a speed of 8½ knots; the fuel consumption was measured during 10 hours, with results as follows:

'Gas Tug No. 1': 530 lbs. German anthracite.
'Elfrieda': 1820 lbs. steam coal.

The steamer thus used nearly 3½ times as much fuel as the suction gas boat.

H.M. Gunboat 'Rattler,' 715 tons displacement, 165 ft. long by 29 ft. beam, built in 1883 as a steam auxiliary, and fitted with horizontal triple-expansion engines, was fitted up by Messrs. Beardmore in 1908 with an experimental Beardmore-Capitaine five-cylinder, four-cycle, single-acting, 20 ins. × 24 ins., 500 HP suction gas engine, running at 120 revolutions per minute; ignition was by low-tension magneto, and the inlet and exhaust valves were water cooled. The gas-making plant comprised an anthracite generator, cooling tower, and centrifugal dryer; the steam for the generator was at first supplied by a donkey boiler; later the exhaust silencers were fitted with groups of tubes through which water was passed, the steam thus obtained being used in the generator. Experiment showed that the engines could be kept running for 20 hours by filling up the generator, but normally it was usual to supply about 200 lbs. of anthracite per hour through the fuel hopper; the average consumption was stated to be 0·8 lb. of fuel per horse-power hour, and the boat maintained a mean rate of 10 knots.

Starting was effected by introducing and firing an explosive mixture of gas and air at a pressure of 100 lbs. per sq. in. into the engine cylinders; reversal caused some difficulty, the engine being uni-directional; at first a device involving compressed air supplied by two Westinghouse air pumps of locomotive type was employed, but this was replaced later by an hydraulic clutch with epicyclic reversing gear. It was hoped that it would be possible to replace the anthracite producer by one of the bituminous type, but the design of the bituminous producer had not then been sufficiently perfected to enable it to be used on shipboard.

H.M.S. 'Rattler' thus fitted was used by the Royal Naval Volunteers under the command of the Marquis of Graham for two months in 1908, in which time a distance of 1750 miles was run on the west coast of Scotland; subsequently many trial trips were made with

Royal Naval officers and others on board, and in April 1910 the boat proceeded to Portsmouth in order that instruction might be given to naval stokers in the working of suction gas engines.

The great difficulty with marine engines of the internal-combustion type is that of efficient manœuvring; in order to further study and perfect the marine suction gas engine the Holzapfel Marine Gas Power Syndicate early in 1910 placed an order for a small sea-going cargo vessel to be used in the South Wales coal trade, with Messrs. Eltringham & Co., of South Shields. The boat 'Holzapfel I' is 120 ft. × 22 ft. × 11 ft. 6 ins., and is fitted with a high-speed, six-cylinder, vertical suction gas engine developing 180 HP at 450 revolutions per minute, built by Messrs. Hindley of Bourton. The engines are placed right aft, and are supplied with gas by a duplicate set of Mond producers, each of 100 HP capacity, constructed by the Power Gas Corporation, Ltd., of Stockton-on-Tees. The producers are of square section, placed together on the port side of the boat and fronting towards the engine room; the scrubbers, about 13 ft high, are placed forward of the producers. Both producers and scrubbers are contained in a gas-tight chamber separated from the engine room, and furnished with two 12 in. ventilators. The power is transmitted to the propeller shaft by means of a 'transformer,' the invention of Dr. H. Fottinger, which in effect is an hydraulic clutch, and consists of a centrifugal pump delivering water into a turbine. There are two turbines, one driving ahead, the other astern. The water is conveyed to them through a reversing valve actuated by a hand lever, and when this lever is in its middle position, the system is in the condition called 'free engine'—that is to say, the engine continues to revolve without driving the propeller. The engine always runs in the same direction, whether the propeller is going ahead or astern. In this vessel the transformer is geared from 450 r.p.m. of the gas engine to 120 r.p.m. of the propeller. The transformer is coupled direct to the engine, and together they occupy a total length of about 20 ft. A donkey boiler, for working the steam winches, and its coal bunker are placed on the starboard side, and together they form a counterbalance for the weight of the gas plant. The bunker to supply the gas producer is in the poop, will hold about 12 tons of anthracite, and will deliver into the hoppers of the two producers, very little trimming being required.

The engine is started by compressed air stored in three separate cylinders at a pressure of about 500 lbs. per sq. in.; these cylinders are charged by two air pumps, one of which is driven by steam from the donkey boiler, the other by a small horizontal gas engine using gas from the producers. During trials of this boat made early in 1911 a speed of 7·6 knots was attained, and the Fottinger hydraulic transmission gear is said to have given satisfaction.

In land installations suction gas plants are now very numerous, large, and successful; bituminous producers are being gradually perfected, and when this is done the problem of the power gas plant may be regarded as fully solved on the large scale. The difficulties arising with bituminous coal have proved very great; such coal may contain from 12 per cent. to 30 per cent. of volatile matter, and may yield as much as 300 lbs. of tar per ton gasified. In a land installation a plant may be fitted for dealing with this and other valuable by-products, but on ship-board it becomes a very difficult problem, the solution of which is still being sought; the extended use of power gas for purposes of marine propulsion cannot be considered as practicable until producers are available capable of supplying clean gas from any grade of bituminous coal. Largely due to this difficulty, and also to that of obtaining effective manœuvring power, great attention is at present being concentrated upon the Diesel type of engine for marine use; in connection with this subject the remarks made by Clerk in his Cantor Lectures, 1905, may be cited; he observed then:

'I now come to the application of producer gas, the suction producer, or any other producer, to sea-going purposes. The sources of petrol are so limited that one cannot expect to do anything great in the way of motive power at sea with petrol. There is not enough petrol, and there is not enough oil, in the world. Any sort of run upon petrol by a huge use of petrol engines would at once put up the price, because the amount of petrol in the world is so small. In fact the amount of the oil is so small, that if it were attempted to run anything like the tonnage of this country with petrol or with oil, it would be impossible to get enough petrol and oil in the world to do it. The consequence is that the real marine problem can only be solved when coal in some form is used on board ship.'

CHAPTER XII

THE FUTURE OF INTERNAL COMBUSTION MOTORS

IT will have been seen from the preceding chapters that the internal combustion engine has already developed in many directions, and that numerous difficulties have been met with, some of which have been entirely and some only partially overcome. In some directions progress has been rapid; in others it has been disappointingly slow. Improvement in thermal efficiency has been rapid; so has application to smaller units; but large power units have presented all the anticipated difficulties, as well as others which were not foreseen. Broadly, however, development has been great and general, and the field of the reciprocating steam engine has been effectively invaded for both stationary and locomotive purposes, both on land and water. The art of flight, too, has at length been rendered possible by the construction of exceedingly light petrol-driven engines of the internal combustion type.

Engineers who have devoted their attention to such motors have amply justified their predictions, and earned an honourable position among the workers of the world; but their task is by no means complete, and an enviable field of effort is still open to the young engineer in aiding the next stage of advance. Many problems present themselves in connection with, for example, thermal efficiency: modification of the composition of the working fluid so as to reduce temperature difficulties: better means of cooling and better methods of construction in order to reduce expansion stresses: better governing cycles so as to divide impulses with greater economy: better means of vaporising oils in petrol and heavy-oil engines: improvements in valve details and igniting arrangements: but perhaps the most pressing of all is that of the elimination of the disadvantages of the large cylinder gas engine, or the replacement of one large cylinder by numerous smaller cylinders, or by some type of rotary turbine, in order to produce units comparable in power to steam turbine units. Then, again, producer-gas problems are pressing: and more work is required on bituminous fuel producers for large powers.

Consider first the question of thermal efficiency. In gas engines the compression ratio commonly used varies between 4·5 and (about) 7, depending on the nature of the gas consumed, and in the Diesel engine it is from 12 to 14. The effective thermal indicated efficiency of constant-volume engines with the higher compression is about 35 per cent., and the corresponding brake thermal efficiency is 30 per cent. Higher brake efficiencies have been found in special tests, but these values are the highest usually attained in practice. Increased compression cannot greatly change the efficiency, apart from the practical troubles attending higher compressions in explosion engines. This becomes very apparent on examining the change of air standard efficiency with increasing compression. With a compression ratio of 5, the air standard efficiency is 0·48, while with 10 it is 0·61, and with 20 only 0·7. The corresponding practical efficiencies in cylinder engines are nearly 0·34, 0·43, and 0·49; taking 0·7 as the efficiency ratio value.

An increase of compression ratio from 5 to 20 thus only increases the possible indicated efficiency from 34 to 49 per cent. Bearing in mind that a compression ratio of 20 is quite beyond the range of an Otto cycle explosion engine, it is obvious that 50 per cent. is a higher value than can be expected from any commercial engine using high compression for improving economy. The values for compressions up to 100 prove that but little further is gained by greatly increasing compression, even in a theoretically perfect air standard engine At 100 compressions the air standard value is 0·85, but to obtain this value the temperature of compression would necessarily rise to about 1600° C., so that no power whatever could be obtained from the engine as an explosion engine without a *further* rise of temperature. No explosion engine, in fact, could be worked at such compression, because the mixture would ignite spontaneously long before the piston reached the end of its stroke. Taking all practical and theoretical matters into consideration, it seems improbable that more than about 40 per cent. indicated thermal efficiency can be attained commercially by raising compression.

There appears to be only one method open to further increase thermal efficiency in a practical way and that is by using regeneration of heat in some form. The utilisation of the waste heat of the water jacket and the heat of the exhaust gases to raise steam has become considerable. It has been already mentioned that about 2·5 lbs. of steam per BHP has been obtained from the exhaust gas heat of large gas engines. Taking 10 lbs. of steam per shaft HP as the consumption of a high efficiency Parsons steam turbine, this would add 25 per cent. to the power of the gas engine without burning any more fuel. A brake thermal efficiency of 30 per cent. would thus be raised to 37·5 per cent.

With an engine giving so high an indicated efficiency as 40 per cent., the exhaust heat would be insufficient in amount to raise 2·5 lbs. of steam per BHP., but if all the waste heat were collected from the jacket, piston valves, and exhaust—amounting to 60 per cent. of the original heat—then the 40 would be increased to nearly 50 per cent. on the assumption that the heat could be utilised in a highly efficient steam turbine. The combined brake thermal efficiency of such a regenerative gas engine would be about 43 per cent.

Raising steam by waste heat is the most practicable of all regenerative methods, but theoretically higher values are possible by dealing with air as the working fluid, and applying the heat of the exhaust to raise its temperature at constant volume after compression and before explosion.

To obtain maximum efficiency by this method, however, it is necessary to adopt the perfect regenerative cycle, which requires isothermal compression at the lowest temperature, and isothermal expansion at the highest temperature. With the usually assumed properties of air (constant specific heat) a perfect regenerator could reduce the working fluid from the highest temperature to the lowest temperature at the end of the expansion, and the heat so stored in the regenerator could raise the fluid from the lowest temperature to the highest temperature at the end of isothermal compression. The whole of the working heat would then be added to the charge during expansion at constant temperature, and all the heat discharged would be discharged at the lower temperature by isothermal compression. Such a cycle would give theoretically the maximum efficiency possible, equal to that of the Carnot cycle. The actual operations required, however, are impossible, from the commercial point of view. In actual practice it would be necessary to use ordinary adiabatic compression, and this would involve a considerable loss in the efficiency of the regenerator. A high thermal efficiency could be obtained, however, with adiabatic compression, and a partial addition of the heat lost with the exhaust, and the heat lost to some of the enclosing surfaces, by the application of the regenerator. In the future, the regenerator will undoubtedly be applied, probably at first in the form of the steam regenerator, but later even air may be heated regeneratively in larger engines with economy.

Although the problem of improving efficiency is a fascinating one from the scientific point of view, it is not at present, however, of vital importance. The present thermal efficiencies are sufficiently good, and it is much more important to improve internal combustion engines in other respects.

In large-cylinder gas engines (i.e. of 30 ins. diameter and above), temperature difficulties are still serious, and are evaded by reducing mean pressures by the use of dilute mixtures, so that the highest practicable mean pressures do not exceed 70 lbs. per sq. in., and even this value can usually only be maintained for short periods without endangering the safety of the engine. Much higher average safe pressures are, however, quite obtainable by the use of super-compression, either with air or exhaust gases. The author has built engines working successfully on both systems. Air super-compression requires an added pump, which absorbs some power, and so reduces the balance of advantage, but exhaust super-compression is not open to this objection, and is much preferable. The object of super-compression is to maintain or increase mean pressure while materially reducing maximum and mean temperatures. The rate of heat flow from the working fluid to the enclosing walls is thus reduced, and in this way the temperature conditions of the cylinder, piston and combustion-chamber surfaces are made less onerous. The temperature stresses are correspondingly reduced. By exhaust super-compression it is easy to maintain mean pressures of 95–100 lbs. per sq. in. accompanied by less severe temperature stresses than those arising with 70 lbs. as ordinarily applied. In a four-cycle engine of 10 ins. diameter × 18 ins. stroke designed for some experiments of this kind, the piston was arranged to overrun a number of small circular ports placed all round the cylinder at the outer end of the stroke. These ports led into an annular space which was carefully water-jacketed. When the piston was near the end of its outer working stroke, it opened these ports, and the exhaust gas at a terminal pressure of 30–40 lbs. per sq. in. above atmosphere thence passed into a reservoir by way of the annular space, and a long pipe cooled either by air or water. A sufficient quantity of the cooled gas was admitted to maintain the reservoir pressure at the desired point of about 5 lbs. per sq. in. above atmosphere.

A positively-actuated piston valve was placed close to the annulus, timed so as to close the pipe and reservoir from the cylinder just before the ordinary exhaust valve of the engine opened. The reservoir was thus charged with exhaust gas by the waste pressure of the exhaust, without interfering with the ordinary discharging operation of the engine, and a quantity of products of combustion was thus obtained at a pressure above that of the atmosphere, without absorbing any work from the engine. At the end of the next stroke (the suction stroke) immediately upon the closing of the charge inlet valve, the piston valve referred to opened the cylinder to the exhaust gas reservoir, and the cooled exhaust gas then flowed by another pipe through the annulus into the cylinder, and so raised the pressure to about 3 lbs. above atmosphere. The weight of total charge present in the cylinder was thus increased, although the inflammable charge remained unchanged in amount. The effect of this addition of non-combustible gas was to diminish both maximum and mean temperatures, while increasing the mean pressure. The increase of mean pressure followed from the application of an unchanged quantity of evolved heat to a greater weight of working fluid, and the increase occurred partly because of the diminution of specific heat of the gases, due to the lower temperatures, and partly because of the diminished heat lost from the working gases to the walls of the cylinder. By this device of super-compression, it is possible to effect considerable economy in gas consumption at full load. In fact, the overload generally specified for large gas engines can be maintained by the use of exhaust super-compression, without the customary fall in thermal efficiency. In the future undoubtedly the composition of the working fluid will be modified by the use of exhaust, or air, super-compression, and by this means many of the temperature difficulties of big cylinders will be largely eliminated. Scavenging by cooled exhaust gases obtained under pressure as described will also aid in keeping down temperatures, and increasing mean pressures.

In existing engines, water-cooling of the cylinders and combustion chambers by jackets operates quite satisfactorily with cylinders of moderate dimensions, but the cooling of pistons by pumping water through them introduces so many complications, and indirectly causes so great an increase in the weight of the engine, that it is desirable if possible to find some other means of keeping their temperature down. Proposals have been made at various times to cool pistons by injecting water into the hollow trunk, but a more interesting method has been ,proposed by Professor Hopkinson, of Cambridge. In this method water is injected into the compression space during the compression stroke of the engine, and nozzles or apertures are arranged so that the water jet sprays against the surfaces of the combustion chamber, the valves, and the piston head. The injected water spray directly removes heat from these surfaces by evaporation. By this arrangement, Professor Hopkinson withdraws the heat of the metal from the interior surfaces, and cools the circular end of the piston from the centre outwards. He dispenses entirely with water-jacketing, either for the cylinder barrel itself or the combustion chamber. Heat thus flows from the cylinder barrel to the piston, and the cylinder walls are kept cool by contact with the piston, instead of cooling the piston in the usual manner. By this arrangement, expansion difficulties between piston end and cylinder are avoided. The temperature of the barrel is arranged to be a little higher than usual with a water-jacket arrangement. This method of internal cooling has been found to operate very well with a Crossley engine, 40 HP, of 11½ ins. cylinder; with a National engine, of 18 ins.

cylinder diameter; and with a large gas engine having a cylinder over 30 ins. diameter. The method has the great mechanical advantage of avoiding jackets altogether. It thus permits very simple castings to be made, and appears, when thoroughly worked out, to be a valuable means of diminishing cylinder and combustion head difficulties in large-cylinder gas engines. It seems likely that it may be used considerably in the future for such engines. By the use of exhaust super-compression, however, and efficiently conducting pistons, it is practicable, as has been pointed out already in this work, to run pistons effectively without water of any kind in the piston, up to about 26 ins. diameter. The future will no doubt see many further detail improvements tending to reduce the temperature difficulties of large-cylinder engines.

Governing cycles have been greatly improved, and in the modern large gas engine consecutive impulses are obtained at less than half the maximum mean pressure; but even yet it is not possible to subdivide impulses so as to get, say, one-twentieth of the maximum mean pressure with certainty and stability. This difficulty also applies to the Diesel type of engine. In Diesel marine engines, considerable reductions can be made in the power developed per impulse, but the great variation of mean pressure obtainable in the reciprocating steam engine has not yet been attained in any gas-engine cylinder.

Flame-injection methods have been tried by the author, and he has succeeded in getting very low average pressures with complete certainty. In future governing cycles, it is probable that even in explosion engines the lower governed impulses will be obtained by flame injection.

The study of better means of vaporising oils, both in petrol and heavy-oil engines, is attracting much attention at present. In petrol engines, great improvements have been recently made, but the difficulties of using heavier oils in small quick-speed engines of the petrol-engine type have not yet been overcome. A great reward awaits the inventor who succeeds in getting as good results from a petrol engine using kerosene or ‘ paraffin ’ as can be at present obtained with petrol. Many devices are being experimented with to get this better result, and although the problem is difficult, there is good hope that it may be satisfactorily solved. The Diesel method of ignition and combustion has advantages in the case of heavy oils, but it has also some disadvantages which make it advisable to consider modified methods carefully.

Improvements in valve details and igniting arrangements are by no means at an end. The introduction of sleeve and rotary valve engines has proved that the lift valve may even be dispensed with in some circumstances. It is highly improbable, however, that lift valves will ever be entirely displaced. The field will perhaps be shared between lift and sleeve and rotary valves in small engines. So far, such sliding-surface valves have not been applied to large cylinder engines, although some advantages would undoubtedly result from their application. The difficulties, however, are much greater than in small-cylinder engines.

Many other detail problems remain to be worked out, but perhaps the most pressing general problem of all lies in the production of internal combustion motors, capable of being commercially applied in very large power units. For such units, the reciprocating steam engine has been so largely displaced by the steam turbine that the advantages of rotary as compared with reciprocating motion have been demonstrated in the most striking fashion. Undoubtedly the modern development of the steam turbine, displacing, as it has so successfully done, all the larger reciprocating power units, and the difficulties of the large-cylinder gas engine, supply an ever-increasing stimulus to engineers desiring to construct internal combustion engines in very large units. Engineers engaged in the development of the internal combustion engine, like steam engineers, have long recognised that great advantages would result from the substitution of rotary for reciprocating motion ; accordingly many attempts have been made to produce a commercial gas turbine. So far, however, no attempt has succeeded ; the practical difficulties have proved too numerous and serious. In connection with gas turbine attempts, the name of the late M. René Armengaud deserves highly honourable mention. He succeeded in getting 300 BHP from a large constant-pressure gas turbine, but to do this he had to reduce the flame temperature of the issuing working fluid to about 400° C., by the addition of steam. His turbine was in fact really a steam turbine using very high superheat. M. Armengaud, however, demonstrated that a gas turbine with turbine compressor gave power sufficient to drive the compression pump, and leave an equal power available at the shaft. Petrol was the fuel used, but the consumption was very heavy. It amounted to about 3 lbs. of petrol per BHP hour. It is to be feared that the constant-pressure turbine presents too many difficulties for success to be hoped for in practice.

An interesting explosion turbine was designed by M. Karovodine, in which explosions from atmospheric pressure were caused to succeed each other very rapidly, and they propelled a small turbine by means of jets. Here, too, the consumption was high. The machine, however, was very small, giving only 1·6 BHP at 10,000 revolutions per minute. Mr. Holzwarth has also experimented with a turbine operating on the explosion principle. In a machine, however, built for a rated power of 1000 horse, he only appears to have succeeded in getting about 160 BHP as a maximum.

In an address to the Junior Institution of Engineers in 1905, the author pointed out the practical difficulties of gas turbines, and these difficulties have not yet been overcome. The necessity of increase in the power unit, however, is so pressing that further experimenting on turbine lines is most desirable. In the author's opinion, very large units of power, such as 20,000 horse on a single shaft, may perhaps be rendered commercially possible for internal combustion engines by the use of combined explosion and water turbines. By communicating velocity to water by explosion pressure, and directing the water against a wheel, for example of the Pelton type, it is possible to avoid all temperature difficulties in the turbine part, and to attain great economies of expansion in the combustion-chamber part. Mr. Humphrey's important work in connection with pumping by the direct action of explosion will doubtless be much further extended, and it appears to the author that the future may witness very large power internal combustion engines in which piston, crank, and connecting-rod are entirely dispensed with, a comparatively light device being employed to cause gaseous explosions to propel water jets, in combination with a turbine to convert the jet velocity into continuous rotary motion. Such an arrangement, it is true, still involves a reciprocating mass of water, but it will probably be found that a great saving in weight will result from the omission of the piston, connecting-rod, engine frame, and crank. This line of investigation appears to the author more hopeful than any scheme involving the direct contact of flame with turbine blades.

The Diesel engine has provided gas engineers with a much-needed stimulus in the direction of applying internal combustion engines to marine purposes, locomotives on land, and the larger engines required for stationary purposes; but although the Diesel engine solves some difficulties, it introduces others, and it does not avoid the limit imposed upon large power units by the increase in weight due to increase of cylinder diameter. This increase in weight has been pointed out in this work, and it acts much more severely in the case of gas engines than steam engines. In the Diesel engine, under similar circumstances, the weight increases more rapidly with increase of cylinder diameter than even in the ordinary explosion gas engine. Accordingly it appears to the author somewhat improbable that the Diesel engine will furnish very large power units. Dr. Diesel and other able German and English engineers who are co-operating with him are attempting large-cylinder Diesel engines, and no doubt they will succeed in running engines with larger cylinders than are at present built; but in the author's view it will be necessary to abandon the cylinder and piston, if internal combustion engineers wish to succeed in producing such large power units as now made, for example, by the Hon. Sir Charles Parsons with the steam turbine. The greatest power which the world has so far attempted to produce on a single shaft is to be found in the steam turbine now being built at Messrs. Parsons' works, and the power is 40,000 horse. Such huge powers will no doubt be possible for internal combustion engines in the future; but to attain them it will be necessary to radically depart from the long-established and at present universally adopted reciprocating type of engine. Finally, it is very desirable that close attention should continue to be given to the further development of the Bituminous Coal Gas Producer. Coal is by far the cheapest and most abundant fuel, and notwithstanding the great developments in oil production, coal of some kind must always remain the principal ultimate source of energy for all types of engine. Oil has a great field of usefulness, no doubt, but will not be able to carry out the extensive programme as predicted by Dr. Diesel. It is particularly to the interest of Great Britain that the Bituminous Fuel Producer should be fully developed, so as to become applicable to marine as well as other internal combustion engines, on the largest scale.

APPENDIX

ON THE ACCELERATION OF THE RECIPROCATING PARTS

Acceleration of Reciprocating Parts.—Many constructions for determining the acceleration of the piston of a reciprocating engine have been given. One of the earliest methods employed (Proëll's) depended upon the fact that the subnormal of the velocity curve is a measure of the acceleration; practically, however, this was useless to the draughtsman. Among positive geometrical methods may be mentioned those of Rittershaus, Klein, Mohr, Kirsch, and Bennett; the subject has also been treated by Professors Unwin, Hill, Ewing, Dalby, and Minchin, and by G. A. Burls (v. *Proc. Inst. C.E.*, vols. cxxiv *et seq.*). One of the best constructions is that of Professor Klein, as originally given by him in the *Journal of the Franklin Institute*, vol. cxxxii, for the general case of a quadric crank chain. For the particular case of the crank and connecting-rod combination it reduces to the simple process described later on in this note.

In the accompanying fig. I, OP, PM represent the crank and connecting-rod respectively, in any configuration.

AM $= x$ (feet) is the corresponding portion of the out-stroke described by the piston. Denote OP by ρ (feet); PM by l (feet); MOP by θ (radians); and OMP by ϕ (radians). Let v denote the arcual velocity of the crankpin in feet per second, and ω its angular velocity in radians per second. Then $\omega = \frac{d\theta}{dt}$, and $v = \omega\rho = \rho \cdot \frac{d\theta}{dt}$; it is usual to consider that v, and hence ω, and $\frac{d\theta}{dt}$ are constant.

Firstly let the problem be considered analytically. We have:

$$x = \text{AM} = \text{AO} - \text{OM} = \text{AO} - (\text{OC} + \text{CM})$$
$$= (\rho + l) - (\rho \cos \theta + l \cos \phi)$$

That is—

$$x = \rho(1 - \cos \theta) + l(1 - \cos \phi) \qquad (1)$$

showing that x is made up of the displacement due to the crank obliquity expressed by $\rho(1 - \cos \theta)$, augmented by that due to the connecting-rod obliquity expressed by $l(1 - \cos \phi)$.

The velocity of the piston is $\frac{dx}{dt}$; differentiating (1) all across with respect to t we have:

$$\frac{dx}{dt} = \rho \sin \theta \cdot \frac{d\theta}{dt} + l \sin \phi \cdot \frac{d\phi}{dt} \qquad (2)$$

showing that the piston velocity is the sum of the components, along the line of stroke, of the crank-pin velocity $\rho\cdot\frac{d\theta}{dt}$, and the arcual velocity of M due to its swing around P, viz., $l\frac{d\phi}{dt}$.

Again, the acceleration of M is $\frac{d^2x}{dt^2}$; differentiating (2) therefore, with respect to the time we have:

$$\frac{d^2x}{dt^2} = \rho\cos\theta\cdot\left(\frac{d\theta}{dt}\right)^2 + \rho\sin\theta\cdot\frac{d^2\theta}{dt^2} + l\cos\phi\cdot\left(\frac{d\phi}{dt}\right)^2 + l\sin\phi\cdot\frac{d^2\phi}{dt^2} \qquad (3)$$

Again a symmetrical expression in ρ, θ and l, ϕ, showing, of course—as accelerations obey the Parallelogram Law—that the acceleration of M towards O is made up of:

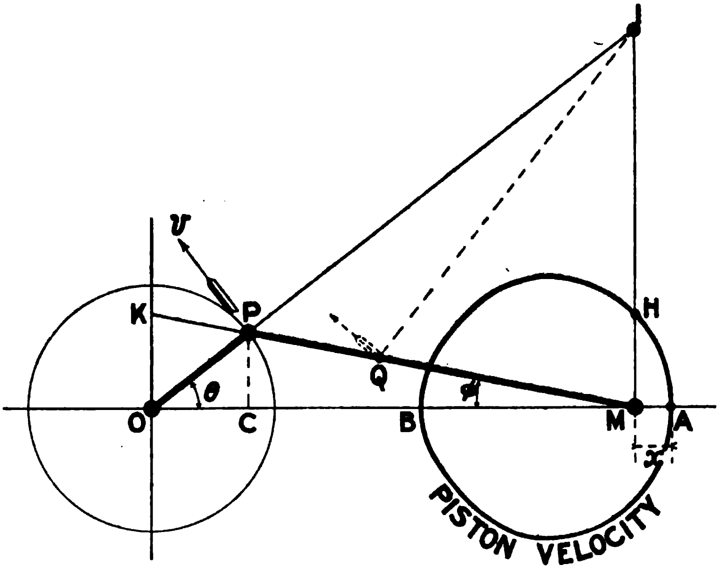

FIG. 1

(1) The component in MO of the centripetal acceleration of P, viz., $\rho\cos\theta\cdot\left(\frac{d\theta}{dt}\right)^2$.

(2) The component in MO of the arcual acceleration of P, viz., $\rho\sin\theta\cdot\frac{d^2\theta}{dt^2}$.

(3) The component in MO of the centripetal acceleration of M towards P due to the swing of the rod around P, viz., $l\cos\phi\cdot\left(\frac{d\phi}{dt}\right)^2$.

(4) The component in MO of the arcual acceleration of M due to its swing around P, viz., $l\sin\phi\cdot\frac{d^2\phi}{dt^2}$.

Now as in (3) $\frac{d\theta}{dt} = \omega$ is assumed constant, we have $\frac{d^2\theta}{dt^2} = 0$, and accordingly this equation reduces to:

$$\frac{d^2x}{dt^2} = \omega^2\rho\cos\theta + l\cos\phi \cdot \left(\frac{d\phi}{dt}\right)^2 + l\sin\phi \cdot \frac{d^2\phi}{dt^2} \qquad (4)$$

In practice the values of the acceleration are usually only required at the ends of the stroke, viz., at A and B. These values may at once be obtained from Eq. (4). For at A, $\theta = 0 = \phi$; hence:

$$\text{Acceleration at A} = \omega^2\rho + l\left(\frac{d\phi}{dt}\right)^2$$

Now we have the geometrical equation $l\sin\phi = \rho\sin\theta$, from which by differentiation, transposition, and squaring, we obtain $\left(\frac{d\phi}{dt}\right)^2 = \frac{\omega^2\rho^2}{l^2}$; substituting in the result just obtained, we have therefore:

$$\text{Acceleration at A} = \omega^2\rho\left(1 + \frac{\rho}{l}\right) \text{ f.s.s.} \qquad (5)$$

Similarly at B, we have $\theta = \pi$, $\phi = 0$, whence from the same result we get:

$$\text{Acceleration at B} = -\ \omega^2\rho\left(1 - \frac{\rho}{l}\right) \text{ f.s.s.} \qquad (6)$$

The minus sign preceding the expression is directional only in its significance, and indicates that at B the acceleration of M is *from* O.

If the connecting-rod were infinitely long the motion of M would be identical with that of C, i.e. would be a S.H.M. and the accelerations at A and B would then be measured by $\pm\omega^2\rho$ respectively, being directed always towards the midde point of the stroke and increasing in proportion to the displacement of M from that point. Hence the effect of the finite connecting-rod is to increase the acceleration of M when at A by the amount $\rho/l \times \omega^2\rho$, and to diminish it by a like amount when at B.

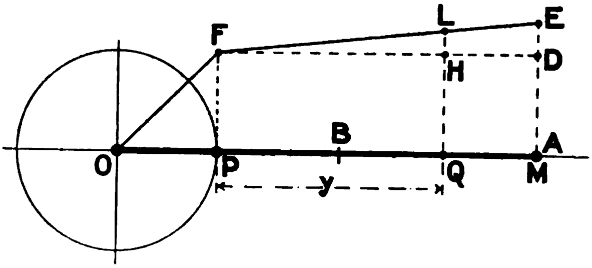

FIG. 2

The results (5) and (6) for the accelerations at A and B can be immediately obtained also in the following manner:

Referring to fig. 2, when M is at A the acceleration of P is $\omega^2\rho$ towards O; also the angular velocity of M around P is momentarily $-\frac{v}{l}$, i.e. $-\frac{\omega\rho}{l}$, and hence the acceleration of M towards P due to this is $\left(-\frac{\omega\rho}{l}\right)^2 \times l = \frac{\omega^2\rho^2}{l}$; hence the total acceleration of M towards O is $\omega^2\rho + \frac{\omega^2\rho^2}{l}$; i.e. as before:

$$\text{Acceleration at A} = \omega^2\rho\left(1 + \frac{\rho}{l}\right) \text{ f.s.s.}$$

Similarly at B the acceleration of P towards O is $-\omega^2\rho$, and of M towards P is $\frac{\omega^2\rho^2}{l}$, and hence the acceleration of M towards O is $-\omega^2\rho + \frac{\omega^2\rho^2}{l}$, or:

$$\text{Acceleration at B} = -\omega^2\rho\left(1 - \frac{\rho}{l}\right) \text{ f.s.s. agreeably with Eq. (6).}$$

Considering further the acceleration at A, we see that the acceleration of any point in the connecting-rod, as Q, distant y feet from P is expressed by $\omega^2\rho + \frac{\omega^2\rho^2}{l^2}\cdot y$:

i.e. $$\text{Acceleration of Q towards O} = \omega^2\rho\left(1 + \frac{\rho}{l}\cdot\frac{y}{l}\right) \tag{7}$$

At P erect a perpendicular PF of length equal to ρ, and let this represent the arcual velocity, v, of P; then as the acceleration of P towards O is $\frac{v^2}{\rho}$, PF also measures to the same scale, the acceleration of P towards O, and thus the velocity and acceleration scales will be alike. At M erect a perpendicular ME, equal, to scale, to $\omega^2\rho\left(1 + \frac{\rho}{l}\right)$, so that $\text{MD} = \omega^2\rho$ and $\text{DE} = \frac{\rho}{l} \times \omega^2\rho$. Join EF and FO; then EFO is the acceleration curve for the connecting-rod and crank in the configuration shown in fig. 2. At any intermediate point in the rod, as Q, the acceleration along the line of stroke is measured to scale by QL.

The necessary force to produce this acceleration could be accurately expressed if the mass M at any point of the rod Q, distant y (feet) from P, could be expressed as a function of y. Thus if at y the mass were at the rate of M lbs. per foot run, then the mass of an element of length at y would be M . dy lbs. The force, $d\text{F}$, in lbs.-weight necessary to produce in this the acceleration $\overline{\text{QL}}$ would be $\frac{1}{g}\cdot \text{M}\cdot dy\cdot\overline{\text{QL}}$. But, by (7) $\overline{\text{QL}} = \omega^2\rho\left(1 + \frac{\rho}{l}\cdot\frac{y}{l}\right)$; hence we have the differential relation:

$$d\text{F} = \frac{1}{g}\cdot \text{M}\cdot\omega^2\rho\left(1 + \frac{\rho}{l}\cdot\frac{y}{l}\right)\cdot dy$$

whence for the whole mass of the reciprocating parts (piston and connecting-rod), when the piston is at the top of its stroke:

$$\text{F} = \frac{\omega^2\rho}{g}\cdot\int_0^l \text{M}\left(1 + \frac{\rho}{l}\cdot\frac{y}{l}\right)\cdot dy \text{ lbs.-weight} \tag{8}$$

In practice this equation cannot be used because M is not expressible as a function of y; from the mode in which it is established, however, the general distribution of the force of acceleration can be mentally realised. Thus as, roughly, in small petrol engines the mass of the piston is about equal to that of the connecting-rod, it is clear that in general for such cases rather more than one-half of the total force is necessary for its acceleration; again about three-fourths of the mass of the connecting-rod is frequently concentrated in the big end; hence about three-eighths of the total force is required for the big end acceleration, the remaining one-eighth serving for the shank and gudgeon eye.

Secondly, referring again to fig. 1, we may with advantage consider the problem of acceleration geometrically.

In fig. 1 produce OP to meet a perpendicular to the line of stroke through M in the point I.

Then, in the configuration shown, P is moving at right angles to IP, and M at right angles to IM ; hence I is the instantaneous centre of the connecting-rod, i.e. this rod is momentarily moving with a motion of revolution about I. Note here that the direction and magnitude of the velocity of *any* point, as Q, in the rod are at once known in terms of the velocity of P. For join IQ ; the motion of Q is at right angles to IQ, and its magnitude is $\frac{IQ}{IP} \times v$.

Thus also, the velocity of M, viz. $\frac{dx}{dt}$, is known, for we have :

$$\frac{dx}{dt}\Big/v = IM/IP \tag{9}$$

Now produce MP to meet a perpendicular to the line of stroke through O in K ; then by similar triangles OPK , IPM, we have :

$$\frac{IM}{IP} = \frac{OK}{OP} = \frac{OK}{\rho}$$

Hence, from Eq. (9) :

$$\text{Velocity of M} = \frac{dx}{dt} = \frac{v}{\rho} \times \overline{OK} \tag{10}$$

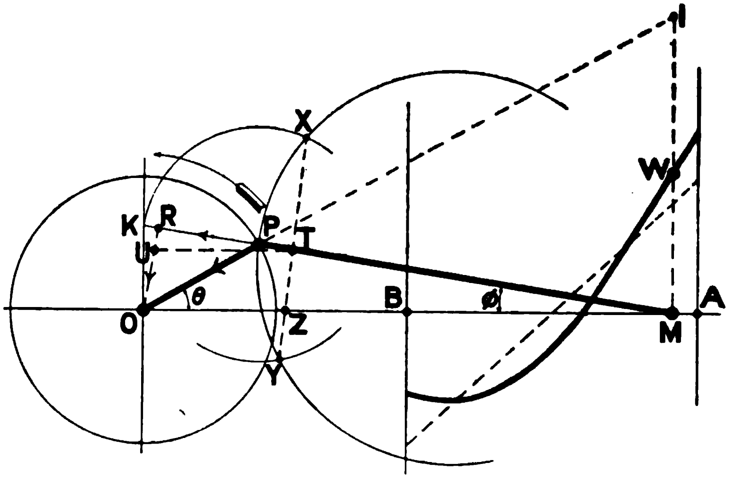

FIG. 3

Thus OK measures the velocity of M to the scale in which ρ measures the arcual velocity of P.

If from M along MI a length MH be taken equal to OK, then the locus of H is the curve of piston velocity.

Acceleration of M *towards* O.—Referring to fig. 3, the acceleration of M towards O is the resultant of :

(1) Its acceleration towards P; and
(2) Its acceleration at right angles to MP.

Now as the arcual velocity of P is assumed constant, the only acceleration of P is $\omega^2\rho$ towards O; and, as accelerations obey the Parallelogram Law, if from O we draw OR at right angles to MK, then, to the scale for which $v = \rho$, PR represents the component of P's acceleration in the direction MP, while RO, to the same scale, measures the acceleration of P at right angles to MP.

Again, the angular velocity of the connecting-rod is $\frac{v}{\text{IP}}$; but $\frac{v}{\text{IP}} = \frac{v}{\text{PM}} \cdot \frac{\text{PM}}{\text{IP}} = \frac{v}{l} \cdot \frac{\text{PM}}{\text{IP}} = \frac{v}{l} \cdot \frac{\text{PK}}{\rho}$; thus the angular velocity of the connecting-rod is $\frac{v}{\rho} \cdot \frac{\text{PK}}{l}$; due to this the acceleration of M along MP is $\left(\frac{v}{\rho} \cdot \frac{\text{PK}}{l}\right)^2 \times l = \left(\frac{v}{\rho}\right)^2 \cdot \frac{\text{PK}^2}{l}$, and this, to the scale adopted, is represented by $\frac{\text{PK}^2}{l}$.

Hence the whole acceleration of M towards P is represented to scale by $\text{PR} + \frac{\text{PK}^2}{l}$. Take $\text{PT} = \frac{\text{PK}^2}{l}$; then TR measures the whole acceleration of M along MP.

We now know the magnitude of one of the component accelerations of M and the directions of the other two; from T, therefore, draw TU parallel to MO, to meet RO in U. Then TRU is a triangle of accelerations for the point M; TR is the component along MP, RU the component perpendicular to MP, while TU is the resultant along the line of stroke MO. It is this latter that is required. From M along MI set off MW equal to TU; then W is a point on the acceleration curve, and MW measured to the scale in which $\rho = v$ gives the acceleration of the gudgeon M in f.s.s. in the configuration shown. Professor Klein's construction enables us to readily determine PT, and thence TU; it is as follows (fig. 3):

On the connecting-rod as diameter describe a circle; from P as centre with radius PK cut this circle in X and Y; join XY and produce if necessary to cut the line of stroke OA in Z. Then OZ measures, to scale, the acceleration of M. For, let XY cut PM in T; then we have $\text{PT}^2 + \text{TX}^2 = \text{PK}^2$, and also $\text{TX}^2 = \text{PT} \cdot \text{TM}$; hence $\text{PT}^2 + \text{PT} \cdot \text{TM} = \text{PK}^2$, and therefore:

$$\text{PT} = \frac{\text{PK}^2}{\text{PM}} = \frac{\text{PK}^2}{l}$$

Thus Klein's construction determines PT; further, XY is perpendicular to MP and is therefore parallel to OR; hence UZ is a parallelogram, and thus $\text{ZO} = \text{TU}$; thus the construction is proved.

Some methods that have been given fail at the dead points, and involve also the practical drawback of lines intersecting at acute angles. Professor Klein's construction fails nowhere, is simple to use, and enables points on the acceleration curve to be readily determined with considerable accuracy.

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